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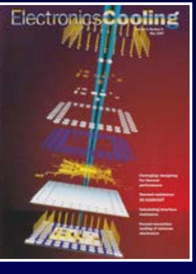

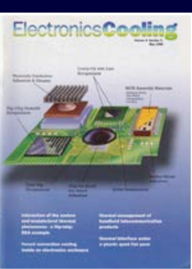

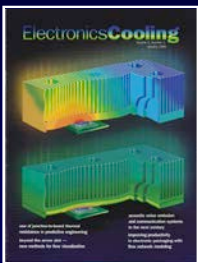

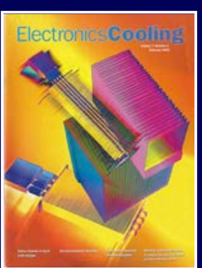

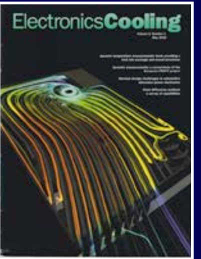

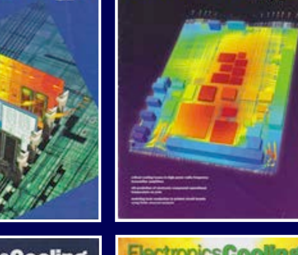

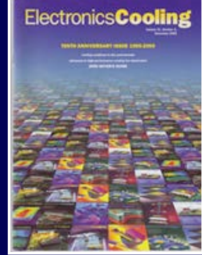

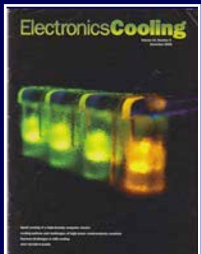

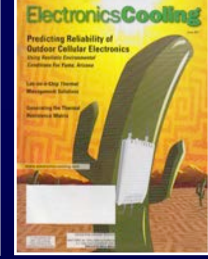

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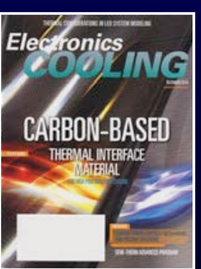

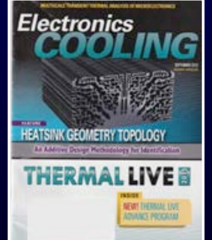

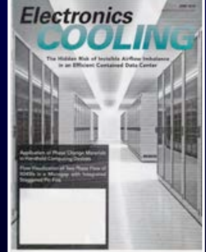

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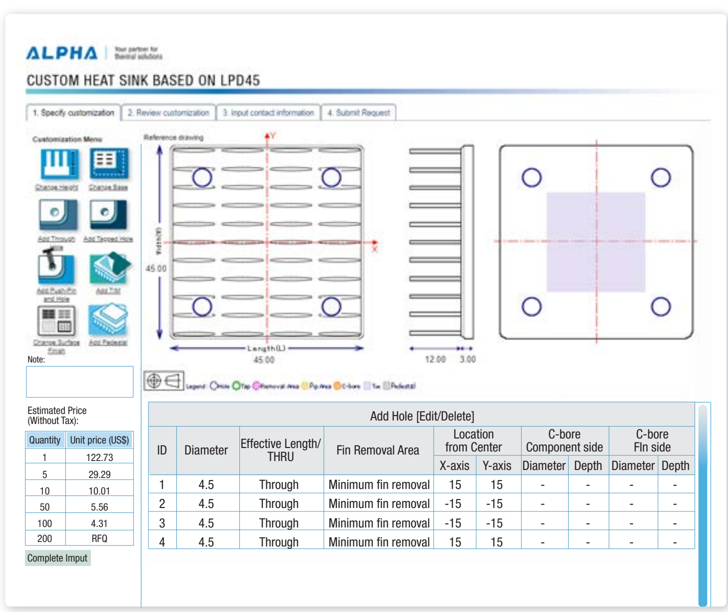

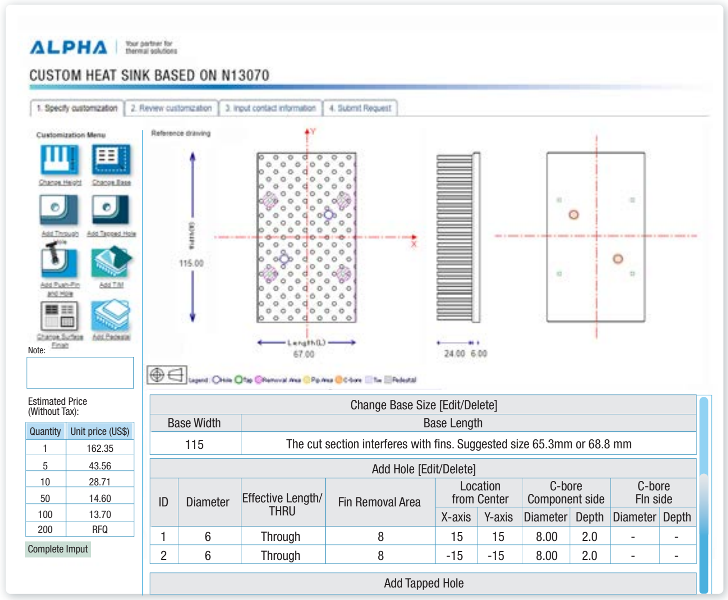

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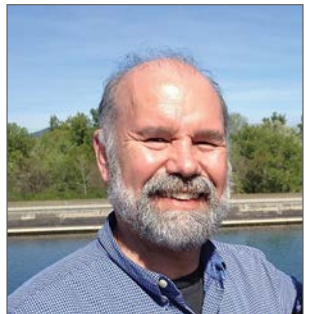
EDITORIAL

Bruce Guenin
Associate Technical Editor



25 YEARS OF ELECTRONICS COOLING

As the "elder statesman" on the Editorial Board of *Electronics Cooling*, it is my sincere pleasure to have been given the opportunity to write this milestone editorial. *Electronics Cooling* has always been of such a scale that its story must be told in terms of the individuals who shaped it at any given time. Memory can be selective, so I apologize in advance for any distortions or omissions in my retelling of this saga.

The first issue of *Electronics Cooling* was published in June 1995, with a publication schedule of three issues per year. The fact that *Electronics Cooling* came into existence at all, is due to the vision and dynamism of a single person. That person is Kaveh Azar, who, at the time, was at ATT Bell Labs in Massachusetts. Contemporaneously, he had founded the company, Advanced Thermal Solutions, Inc. He was aware that in many small-to-midsize electronics companies, there would be a thermal engineer or two and a test technician. He also knew that it was quite common for the thermal engineer to get assigned to the development of product in its very last stages. This was due to the assumption of management that any shortfall in thermal performance could simply be solved by specifying a more efficient heat sink or a more powerful fan. (Unfortunately, as we well know, life is rarely so simple.)

Thermal engineers tended to feel isolated in their own organization and lacked a network of thermal engineers beyond their company to help them expand their skills or knowledge base. He realized that the worldwide community of thermal engineers needed a centralized source of practical information and know-how to keep up with the rapidly increasing demands of the electronics industry. The core vision he had for *Electronics Cooling* was that it would offer content such as technical briefs (describing test or simulation techniques), information on new products (including software, hardware, and materials), news briefs (highlighting conferences, publications, patents, etc.) and in-depth technical articles (conveying sufficient detail to the readers that they could apply the results to their own work). In summary, his vision was that *Electronics Cooling* would become a comprehensive source of current and practical thermal management information of archival value.

Due to his entrepreneurial spirit, he convinced others of the significance of this vision, and in short order, had received a commitment from Flomerics to support the publication and distribution of a new magazine—*Electronics Cooling*. He attracted two of the leading thermal engineers of the day to join this enterprise as Associate Editors: Gordon Ellison, author of the first text book on heat transfer devoted entirely to electronic systems, and Clemens Lasance, of Philips Research in Eindhoven, who had achieved a prominent profile in the international thermal community as he campaigned for greater rigor in thermal engineering and in the calculation of component reliability, as influenced by their temperature.

Electronics Cooling very quickly became a forum that enabled the leading thermal engineers of the day to share their work with its diverse readership. In fact, Kaveh's vision was so compelling that, for those who either came on board as an editor or served as a frequent contributor, their participation invariably became a labor of love. Kaveh had long wanted to have a regular column that he had named "Calculation Corner." I first got his attention as a result of my presenting an evening tutorial at SEMI-THERM® dealing with spreadsheet methods of thermal calculations. In 1998, he approached me with the proposition that I author the column, and I agreed. The first installment of "Calculation Corner" was published in the September 1998, issue. The column has been a regular feature in *Electronics Cooling* ever since.

Shortly thereafter we were joined by Bob Simons, who had had a distinguished career at IBM in the thermal engineering of mainframe computers, and Jim Wilson, an engineering fellow at Texas Instruments, and an expert in the thermal engineering of defense electronic systems. With Kaveh at the helm and with four associate editors on staff, in the year 2000, *Electronics Cooling* began publishing four issues per year. Bob and I shared the "Calculation Corner" column and Clemens and Jim authored the "Technical Data" column. In time, as they were running out of relevant technical data to publish, Clemens proposed a new column, "Thermal Facts and Fairy Tales," devoted to exposing misconceptions and faulty practices in the industry, that were accepted as valid by numerous engineers. Jim alternated with Clemens in authoring this column.

It was a good time for *Electronics Cooling*—the economy was good with strong advertising revenues and equally important, thermal engineers still had enough discretionary time that they could commit to writing up their work for publication. The "lean and mean" era was still in the future. We even had the luxury of having Tony Kordyban (author of the book, *Hot Air Rises and Heat Sinks*) as an occasional guest contributor, as he inserted his dry and quirky humor into investigations of common yet questionable practices in the industry. This "smooth sailing" era came to an end in 2007 when Kaveh left *Electronics Cooling* to devote more time to his other enterprises. In 2007-2008, the financial crisis happened and Flomerics was acquired by a larger company and was compelled to end their financial support for the magazine. More broadly, the crisis put financial stress on all industries, including publishing.

After a period of uncertainty, *Electronics Cooling* was acquired by ITEM Media in 2010. The editors were very relieved, since there was a real risk that the publication would go out of existence. In fact, in the Spring 2010, issue, our first issue after the regime change, Jim Wilson's editorial was headlined, "We are back—and in print." ITEM took on the challenge of creating a sustainable business model around *Electronics Cooling*, while at the same time maintaining the editorial independence of the Assoc. Technical Editors, now referred to as the Editorial Board. Along the way Clemens and Bob left the magazine. Madhu Iyengar, of IBM at the time, and Prof. Peter Rodgers, then at the Petroleum Institute, in the U.A.E, replaced them on the magazine. Madhu began contributing to "Calculation Corner" column and Peter likewise to the "Thermal Facts and Fairy Tales" column.

Meanwhile, structural changes continued in the tech industry worldwide. The tech sector in the west saw a lot of mergers, acquisitions, restructuring events, and layoffs. In this relentless economic environment, as would be expected, time pressures on engineers increased markedly. However, I'm glad to say that, despite these challenges, the editors were able to solicit articles on a wide variety of topics that met the traditional quality standards of the publication. However, it took a lot more effort than in the past to locate authors who could find the time to craft a quality article on a compelling topic.

In 2014, Madhu joined one of the internet giant companies and had to leave the magazine. By 2016, Peter also left to focus all of his energies on the demands of his academic career. In 2016, Jim resigned after being a valued contributor to *Electronics Cooling* for 18 years. Later in 2016, Ross Wilcoxon, then a principal mechanical engineer at Rockwell Collins, and Victor Chiriac, then the thermal technologist at Qualcomm, joined the magazine. Ross has contributed to the "Thermal Facts and Fairy Tales" and "Calculation Corner" columns and has recently started a new column, "Statistics Corner." Victor launched the column, "Technology Corner," focusing on the megatrends in the tech sector with a focus on the semiconductor industry. In 2017, the number of issues per year was reduced to three. In 2019, Genevieve Martin joined us. She is an R&D manager at Signify (formerly a lighting division of Philips), in The Netherlands. She is active in several EU conferences and consortia dealing with LED test standards. Once again, there was a full quota of technical editors, whose expertise covered a wide swath of relevant technologies.

We owe a debt of gratitude to the upper management of Lectrix (formerly ITEM Media) Graham Kilshaw (CEO) and Geoffrey Forman (VP of Marketing) for their role in acquiring *Electronics Cooling* in 2010, and blending it into their print assets as well as giving it a much more robust online presence, thereby integrating it into a sustainable business model. All this was done while retaining their commitment to maintaining the quality standards that have distinguished *Electronics Cooling* over the decades. Without this intervention, *Electronics Cooling* would not now be celebrating its 25th anniversary.

We also applaud their hiring for the first time in 2019, a full time Editorial Director, Jennifer Arroyo, to oversee all the print and online content. She is a seasoned professional, having served as Editor-in-Chief of *ECN (Electronic Component News)*.

Finally, I would like to express our gratitude to you, our loyal readers, for your support over the decades. Without your support, this adventure would not have been possible.

The saga of the ups and downs and comings and goings of the folks associated with *Electronics Cooling* is a microcosm of the wider world around us. However, despite all of these challenges we are still motivated by the original vision of *Electronics Cooling*, that at its very core was seen as a means of bringing engineers today to share their expertise with each other to enable them to be more effective in their professions and, ultimately, to better serve our larger society.

Speaking of goings, I've decided that the time has come for me for me to recede into the background, after spending 22 rewarding and challenging years as an Assoc. Technical Editor of *Electronics Cooling*. I have felt both privileged and humbled by having the opportunity to serve you for so many years. I still hope to provide behind-the-scenes assistance to Ross, Victor, and Genevieve, as they would wish and as I am able. Meanwhile, I pass the torch to another generation of editors and wish them the best of success in these very challenging times.

In this editorial, I've spoken of the challenges faced by *Electronics Cooling*, that were mainly economic in origin. However, we are now facing the most dire challenge in our lifetimes—the COVID-19 pandemic.

At this juncture, no one knows when the world will get this pandemic under control and what its ultimate impact will be on ourselves, our families, and our society.

All I can do, in this final editorial of mine, is to convey my deepest regards to you and your families and my sincere wish that you all stay safe and healthy.

Now, I'd like to add a few words about the content in this special 25th anniversary issue. In honor of this event we have deviated from our usual format and have reprinted articles that were individually selected by each of the Associate Technical Editors from the top 100 articles ranked according to reader popularity. The relative popularity of all the articles at the *Electronics Cooling* website was determined by Google on the basis of the number of unique page views of each article. We hope you enjoy them.

PAST AND PRESENT TECHNICAL EDITORS

TRIBUTE TO BRUCE GUENIN'S 22 YEARS OF SERVICE FOR *ELECTRONICS COOLING* MAGAZINE

On behalf of the past and present members of the Technical Editorial Board of *Electronics Cooling*, it is a pleasure to write a few words of thanks to Bruce Guenin for his 22 years of service to the publication. As some of you may know by now, Bruce has decided to step down from his current EC responsibilities in order to pursue other exciting activities. We wish Bruce all the best in his future endeavors!

To date, Bruce is the longest serving Associate Technical Editor of *Electronics Cooling*—his commitment and contribution to the magazine dates back to 1998. Over the decades, Bruce witnessed many changes in the editorial boards and the magazine's ownership, helped overcome various technical and editorial challenges, championed various initiatives, and maintained his outstanding service and loyalty to the thermal community readership. Under his technical stewardship, the publication has grown and the article quality has always been top notch, as Bruce believes in promoting and applying the highest standards for the technical review process prior to publication. It is the practical value to the reader that he pioneered and professed since the early days of *Electronics Cooling*, and we celebrate Bruce for his long-term vision and dedication as he passes on the baton to us, his editorial board peers.

Thank you Bruce, for your outstanding activity, creativity, inspiration, and commitment to high-quality articles and teachings, from the young to the more seasoned practicing engineers, as well as for the overall (global) thermal community. We will continue to keep the *EC* tech tradition alive and contribute to the publication's continued success.

Below are a few short stories and memories from Bruce and from past technical editors of *Electronics Cooling*. Enjoy and join us in thanking Bruce for his outstanding service to our *Electronics Cooling* community! (compiled by Victor Chiriac, Assoc.Tech. Editor on behalf of the current Assoc. Tech. Editorial Board members Genevieve Martin and Ross Wilcoxon)

Some personal notes on Bruce's retirement from *Electronics Cooling*
(Clemens Lasance, EC Tech Editor, 1995 – 2013)

In 1995, Kaveh Azar, Gordon Ellison, and I started the magazine to fill the gap between academic and advertorial-driven journals. Its purpose from the start was to provide the community of thermal engineers and designers with peer-reviewed articles in which mentioning the name of a company was considered a mortal sin. When Gordon stepped down his place was filled by Bob Simons and Bruce, and when Bob retired Jim Wilson became his successor. I have very fond memories of this period, and I learned a lot from my brother-editors. All articles and columns were peer-reviewed, and I don't recall a single situation where we didn't reach consensus, often after some fierce discussions. Especially between Bruce and me, he the wisest, me the more outspoken (being Dutch). From a scientific/technological point of view his contributions over the years (especially his "Calculation Corner") were highly praised, and, in my humble view, should be archived in a volume to facilitate access. Apart from the magazine, we met regularly at various thermal conferences and JEDEC meetings.

A long time ago Bruce and I found out that he played the guitar and me the piano. I suggested that we should perform as a duo at some SEMI-THERM conference, and I sent him the score of a piece by Mauro Giuliani. It never happened (probably due to the lack of a piano in the conference hall). In 2014, Bruce and his wife Anita visited us in our little village near Eindhoven (now world-wide known as the Van Gogh Village), and I vividly recall our long discussions related to religion and philosophy.

I miss you, Bruce, your wisdom and your humor. I resigned after 18 years, around 2013, he beat me by four years; for sure he will be remembered as the King of *Electronics Cooling*. (notes compiled by Clemens Lasance, at the invitation of Genevieve Martin, EC Assoc. Tech Editor)

Some Random Reminiscences from my 22 Years at *Electronics Cooling*
(Bruce Guenin, EC Tech Editor, 1998 – 2020)

I was asked by my fellow Assoc. Tech. Editors to share my reminiscences regarding my years at *Electronics Cooling*. Below are a few anecdotes that bubbled up from my selective memory into my conscious mind. I hope you find them interesting.

By the time I was invited by Kaveh to join the editorial staff of *Electronics Cooling* in 1998, I had already been rather active in publishing and giving presentations at various venues. But what was different here was that, as far as my "Calculation Corner" column was concerned, it was basically my decision what topic would be explored. Thus, the column was a conduit through which I could communicate directly with tens of thousands of readers of the publication—a novel experience for me at the time.

In the beginning, the titles of my "Calculation Corner" columns had bland, technical titles, such as "Determining the Junction Temperature in a Semiconductor Package, Part III—The Use of the Junction-to-Board Thermal Characterization Parameter," published in the May 2002, issue. Over time, I chose titles that were a bit more playful, such as, "The 45° Heat Spreading Angle—An Urban Legend?" in the November 2003 issue, and "Thermal Vias—A Packaging Engineer's Best Friend," in August 2004. These titles were certainly more "catchy" but they at least revealed to the reader what the content was.

However, this playfulness backfired when I explored an interesting phenomenon I discovered though some simulations. They showed that the junction-to-case thermal resistance of a package, that normally is seen as an intrinsic property of the package, could, in fact, be influenced by the heat spreading properties of the test board. (The board's heat spreading ability was related to the presence or absence of copper planes in its interior.) This finding was a significant one and should have been of interest to the readership. However, in my efforts to be clever, I gave the article the title, "A Funny Thing Happened on the Way to the Heatsink," which gave the reader no idea at all what the article was about. To get the humor, one would had to have heard of the Broadway play, *A Funny Thing Happened on the Way to the Forum* (the story was set in Roman times). Of course, few of our readers had heard of the play and would, therefore, would have been able to get the joke. Thus, the ultimate impact of this title was to obfuscate what the article was about and provide no motivation at all to the reader to investigate the article further. This taught me a lesson and, from that point on, I made no more lame attempts at humor in my "Calculation Corner" titles.

After Kaveh left the publication in 2007, we four Assoc. Tech. Editors adopted the organizational model of a string quartet. We essentially made decisions by consensus. One effect is that since we had four issues per year and four editors, we decided that each of us would be an Acting Editor-in-Chief for one of the issues and would have the privilege of writing the Editorial for that particular issue. This was something I enjoyed very much. It was yet another way of communicating directly with our readers, but one in which I could share my point of view on a broader array of issues that had an impact on our profession and, on occasion, on our society as a whole. As with "Calculation Corner" I also had some fun with the titles of my editorials, by mimicking the titles of well-known works that were current at that time, such as:

- "The Audacity of Engineering," February 2009, from: Barack Obama, *The Audacity of Hope*
- "The Joy of Engineering," June 2015, from: *The Joy of Cooking*, a classic cookbook
- "It Takes a Village—To Raise an Engineer," June 2016, from: Hillary Clinton, *It Takes a Village: and Other Lessons Children Teach Us*

I'm sure that there was a bit of wincing on the part of readers familiar with these references. However, these titles probably helped get their attention. I also had fun with two provocative titles for two installments of the "Thermal Facts and Fairy Tales" column:

- "The Junction-to-Case Thermal Resistance: A One-Dimensional Underachiever in a Three-dimensional, Conjugate Heat Transfer World", Spring 2018
- "Whatever Happened to the Predicted Data Center Energy Consumption Apocalypse?" Spring 2019 (I couldn't resist the opportunity to use the word, "apocalypse")

Although I enjoyed writing these editorials and sharing my point of view on some broader issues with the readership, I had no idea to what extent they were actually read. However, I was very pleased to have been approached by an engineer at a conference who said that after reading "The Joy of Engineering" his son, who was about to start college, decided to major in engineering. That one acknowledgement certainly made it all worthwhile for me. (compiled by Bruce Guenin at the invitation of the *EC* Tech Editorial Board)

TECHNICAL EDITORS SPOTLIGHT

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VICTOR CHIRIAC | GLOBAL COOLING TECHNOLOGY GROUP

Associate Technical Editor

A fellow of the American Society of Mechanical Engineers (ASME) since 2014, Dr. Victor Adrian Chiriac is a co-founder and a managing partner with the Global Cooling Technology Group. He previously held technology/engineering leadership roles with Motorola (1999-2010), Qualcomm (2010-2018) and Huawei R&D USA (2018-2019). Dr. Chiriac was elected Chair of the ASME K-16 Electronics Cooling Committee and was elected the Arizona and New Mexico IMAPS Chapter President. He is a leading member of the organizing committees of ASME/InterPack, ASME/ IMECE and IEEE/CPMT ITherm Conferences. He holds 19 US issued patents, 1 US Trade Secret, 1 Defensive Publication, and has published over 109 papers in scientific journals and at international conferences.

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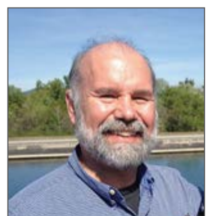

BRUCE GUENIN | CONSULTANT

Associate Technical Editor

Dr. Bruce Guenin has spent many years in the electronics and computer industries, which has given him a broad perspective on macro trends in these fields. He has been an editor of *Electronics Cooling®* since 1997 and has contributed, to date, 35 installments of the tutorial column, Calculation Corner. His previous affiliations include Oracle, Sun Microsystems, and Amkor. He is a past chairman of the JEDEC JC-15 Thermal Standards Committee and the SEMI-THERM® Conference. His contributions to the thermal sciences have been recognized by receiving the Harvey Rosten Award in 2004 and the Thermi Award in 2010 from the SEMI-THERM® Conference. He received the B.S. degree in Physics from Loyola University, New Orleans, and the Ph.D. in Physics from the University of Virginia. He has authored and co-authored over 80 papers and articles in the areas of thermal and stress characterization of microelectronic packages, electrical connectors, solid state physics, and fluid dynamics and has been awarded 18 patents in these areas.

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Genevieve Martin (F) is R&D manager for thermal & mechanics competence at Signify (former Philips Lighting), The Netherlands. She is working in the field of cooling of electronics and thermal management for over twenty years in different application fields. From 2016 to 2019, she coordinates the European project Delphi4LED (3 years project) dealing with multi-domain compact model of LEDs. She served as General chair of SEMI-THERM® conference and is an active reviewer and technical committee in key conferences SEMI-THERM®, Therminic, Eurosime.

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ROSS WILCOXON | COLLINS AEROSPACE

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Dr. Ross Wilcoxon is an Associate Director in the Collins Aerospace Advanced Technology group. He conducts research and supports product development related to component reliability, electronics packaging and thermal management for communication, processing, displays and radars. He has more than 40 journal and conference publications and is an inventor on 30 U.S. Patents. Prior to joining Rockwell Collins (Now Collins Aerospace) in 1998, he was an assistant professor at South Dakota State University.

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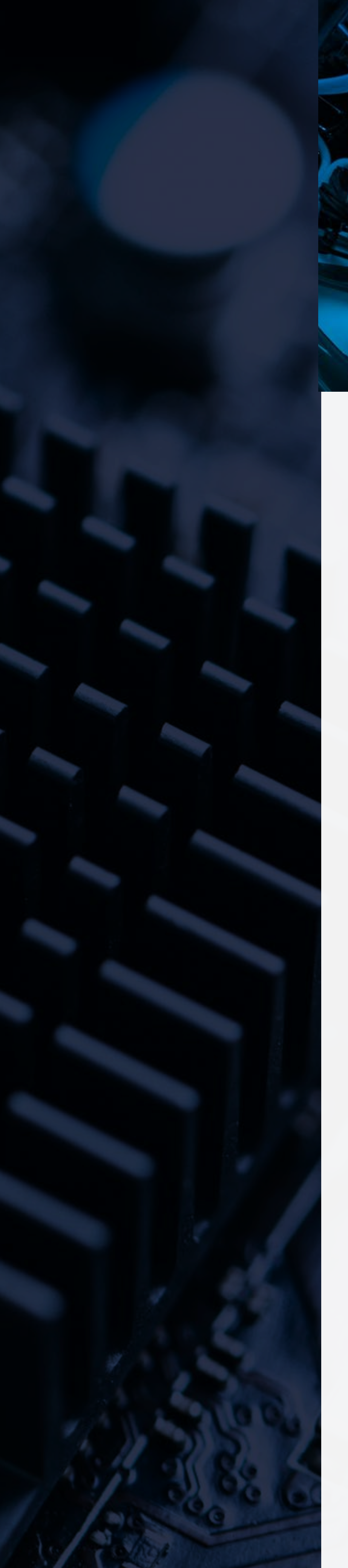

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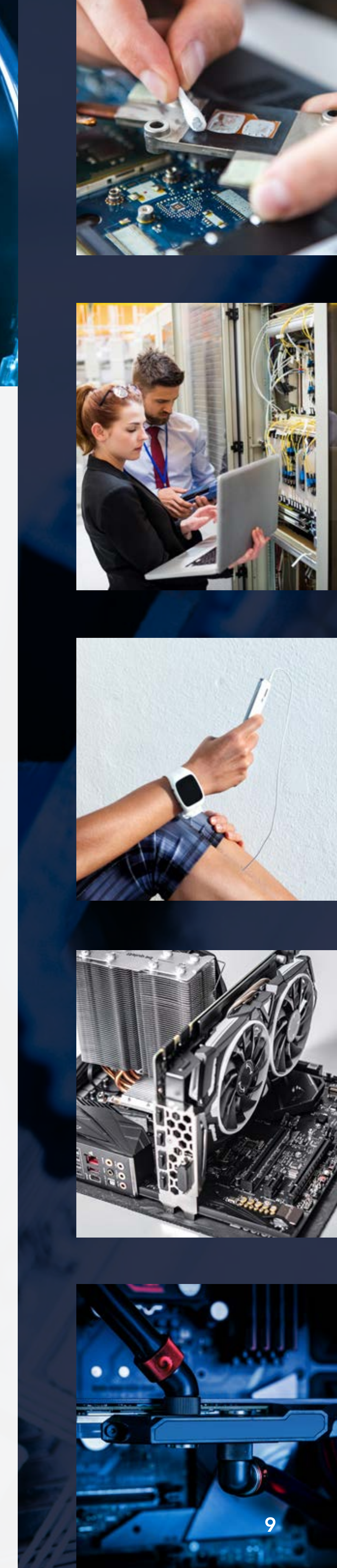

THERMAL LIVE™ 2020 TECHNICAL PROGRAM

OCTOBER 20 – 21, 2020

TUESDAY | OCTOBER 20, 2020

DEREJE AGONAFER, Ph.D., NAE | 9:00 am ET

Presidential Distinguished Professor in MAE, University of Texas at Arlington (UTA)

KEYNOTE - Single Phase Immersion Cooling of Data Centers: Opportunities and Challenges

Data center cooling has never been more challenging as it is today with the thermal design power (TDP) of the components rising by almost 50% in the past decade. With advancements in 2.5D and 3D packaging architectures for realizing smaller feature sizes in this decade, an increase in local power densities has necessitated the requirement of more efficient cooling technologies for thermal management. This growth can be primarily attributed to advancements in high power CPUs and GPUs to support high power computing applications like bitcoin mining, AI, and machine learning algorithms development. Due to low thermal mass limitations, the typical air-cooled data centers present a relatively less efficient and complex cooling option owing to the requirements of raised floors, air handling/chiller piping, power routing, complex filtration and tight environmental control involved, etc. Single-phase immersion provides simplicity in terms of thermal infrastructure, PUEs as low as 1.03, and reduction in CAPEX equal to or greater than 50%. Single-phase immersion can also provide benefits of direct to chip cold plate cooling by eliminating the complexity and improved upfront affordability and ease of operation. Immersion cooling may also prove to be an efficient solution for 3D stacked and heterogeneously packaged dies because of its inherent ability to provide even temperature profiles. Immersion cooling also enhances system reliability by protecting the IT equipment from the harsh environmental effects of high temperature, dust, vibrations, and corrosive gases. Lack of published data on material compatibility, long term signal integrity, changes required in server architecture, and maintainability are cited as some of the primary challenges in large scale implementation. Other major issues associated currently with single-phase immersion are using completely sealed HDDs or using solid-state drives. Additionally, the lack of manufacturer warranties provided by the OEMs is another issue that is slowing the widespread adoption of single-phase immersion cooling. Enhanced equipment reliability and ease of implementation, especially for edge data centers, also make single-phase immersion cooling an attractive alternative for data centers to switch to immersion cooling.

TIM JENSEN | 10:00 am ET

Sr. Product Manager for Engineered Solder Materials, Indium Corporation

Novel TIMs to Enable Advanced Thermal System Design for High Performance and Quantum Computing Applications

Thermal system design is a critical aspect in both high-performance and quantum computing applications. Part of that system is the thermal interface material (TIM). This presentation will begin with an overview of the thermal demands in these applications. From there, the focus will be on unique TIM technologies that can help enable or enhance active and passive cooling designs to ensure maximum performance. Because the use of metal-based TIMs is proving to be increasingly critical for success, several attributes of various metal TIMs will be reviewed to show how they can ensure the best performance in these applications.

THERMAL LIVE™ 2020 TECHNICAL PROGRAM

OCTOBER 20 – 21, 2020

TUESDAY | OCTOBER 20, 2020 | *CONTINUED*

KIMBERLY FIKSE | 11:00 am ET

Lead Sales Engineer, ACT

Advanced Thermal Techniques Across Diverse Industries

In this webinar we will share real world examples where creative approaches in thermal engineering have allowed design engineers to push their design concepts forward. Join us as we explore some advanced thermal management technologies, including Heat Pipes and Phase Change Material heat sinks. We will provide practical applications for each that apply to a range of markets from Spacecraft thermal control to defense and industrial applications.

CHRISTIAN MAINEGRA | 12:00 pm ET

Applications Engineer, Fujipoly

Advantages of Putty Type: Thermal Interface Gap Filler Materials

Fujipoly will reveal many of the advantages of putty-type thermal interface materials offer compared to standard gap filler pads. Key topics will include stress strain comparisons of putties and standard gap fillers, advantages of putty over dispensable thermal interface materials, as well as putty handling and application suggestions.

C. T. KAO | 1:00 pm ET

Product Management Director, Cadence Design Systems

Celsius Thermal Solver: The Thermal Solution for Intelligent System Design

As power densities on IC designs continue to increase, controlling temperature on the chip is becoming a major challenge for IC designers. High temperature impacts both the reliability and electrical performance of ICs, and designers need a way to accurately perform thermal analysis to model the thermal impact of floorplan changes. Furthermore, heat would propagate from the chips through the packages and PCBs to the environment, which clearly indicates that solving thermal issues in modern electronic systems will need to take a global approach but cover the details in the design at the same time. Multi-physics analysis is also essential for system-level thermal solution, including both steady and transient simulation, electrical-thermal co-simulation, and Computational Fluid Dynamics (CFD). Today's advanced applications whether they are in automotive, datacenter, mobile, healthcare or high-performance computing, system-level thermal analysis, which is typically not addressed by other tools, is a critical factor in understanding the overall thermal behavior and managing the thermal impact on performance, reliability, and quality.

THERMAL LIVE™ 2020 TECHNICAL PROGRAM

OCTOBER 20 – 21, 2020

TUESDAY | OCTOBER 20, 2020 | *CONTINUED*

JONATHAN TAYLOR | 1:45 pm ET

Product Manager, Neograf Solutions, LLC

NeoNxGen™ Thermal Management Solutions – Increase the Thermal Performance, Not the Layers!

The latest generations of high-performance consumer electronics need thermal spreaders that are both very thin and highly thermally conductive. One solution is to stack several layers of thin synthetic graphite using layers of adhesive. This method produces spreaders that are thermally effective but are expensive and have poor yields. Layering synthetic graphite creates a final part that is very stiff and can easily delaminate if repeatedly flexed. NeoNxGen™ flexible graphite is a new type of monolayer high-performance graphite heat spreader that is currently being manufactured in high volumes. It has the thermal performance of a multilayer synthetic solution but maintains the flexibility and manufacturing cost advantages of a single layer. NeoNxGen™ flexible graphite is tunable to meet the thickness and performance needs of specific devices, such as the foldable screen in a notebook computer or to replace the copper vapor chamber in a mobile phone.

MICHAEL GONZALEZ | 2:45 pm ET

Sales Engineer, CDCP, CEJN North America

Blind Mate Couplings: Design and Manufacturing Aspects

This session will focus on the increasing liquid cooling demand with required quality, tolerances and technical design considerations of a blind mate coupling, which is needed to ensure a safe and reliable cooling solution for the end user. Effects on manufacturing, QC, and production are also factors that are continually evaluated throughout the design process.

ANAND SAMANT | 3:45 pm ET

Global Market Director for Consumer Electronics, Materion

Advanced Materials Strategies for Cooling Modules in Electronic Systems

Higher data-transfer rates and power-usage requirements result in increasing heat-management challenges for thermal engineers in the electronics industry. At the same time, there is growing demand for thinner and lighter devices, resulting in less available volume for cooling modules. Mismanagement of heat can not only negatively impact short-term performance, but can also lead to failure of electronic components. In this webinar, we will discuss innovative strategies, such as clad systems combining multiple metals, metal matrix composites combining metals and ceramics, and high conductivity monolithic alloys – all of which can help the thermal engineer achieve the desired structural and thermal properties for cooling module materials.

THERMAL LIVE™ 2020 TECHNICAL PROGRAM

OCTOBER 20 – 21, 2020

WEDNESDAY | OCTOBER 21, 2020

BRAD WHITNEY | 10:00 am ET

Product Director/ VP of Two Phased Cooling, Boyd Corporation

Utilizing Ultra-Thin Two-Phase Cooling & Ultra-Thin Titanium Vapor Chambers to Improve Performance

Ultra-thin two-phase technologies such as Vapor Chambers, Titanium Vapor Chambers, and Heat Pipes can be a cost-efficient way to vastly improve the performance of your thermal solution while enabling thinner, lighter cooling. These passive solutions also offer extremely high reliability and lifetimes while remaining silent with no moving parts. These factors make them ideal for more powerful low-profile and next-generation devices. Learn how new technical advancements have enabled improved design and manufacturing options for thinner, higher performance vapor chamber solutions.

ANDRES ABRAHAM | 11:00 am ET

Applications Engineer, Thermal Business Unit, CPC

Liquid Cooling: Getting in the Know About Flow

To optimize performance and enhance sustainability, engineers need to consider how managing fluid flow for cooling within an advanced electronics application differs from traditional air-flow cooling systems. Understanding the heat transfer coefficient and how heat dissipates through liquids exponentially more efficiently than through gases is at the root of efficiencies to be gained. However, understanding flow coefficients alone is not enough. It is only the first step in how to compare liquid cooling component performance or ultimately arrive at system capacity. For example, at the connector level, the Cv value of a given quick disconnect (QD) can vary widely based on the cooling fluid used and system operating temperatures. Also to be considered is volumetric flow rate -- the volume of fluid that passes through a point in a system per unit of time (which is usually expressed in gallons per minute.) Additionally, specific gravity and how fluid properties impact system component requirements can not be ignored.

In this session, learn more about these calculations along with the three design elements within QD design that impact flow. Size, valves, and mounting options are key variable features of liquid cooling connectors, and these components and their interplay or impact on flow is not always intuitively understood. As you will learn, in liquid cooling, efficient flow is essential to effective thermal management.

THERMAL LIVE™ 2020 TECHNICAL PROGRAM

OCTOBER 20 – 21, 2020

WEDNESDAY | OCTOBER 21, 2020 | *CONTINUED*

KEITH PERRIN | 12:00 pm ET

Global Director for Electronics, Hexagon Manufacturing

Advanced Micro-structural & thermal modeling for Energy Generating, Wearable, Electronics

At Hexagon we're dedicated to bringing the virtual and physical worlds together to enable our customers to drive their ideas forward and stay ahead. Nowhere is this more important than in the design and manufacture of industrial and consumer electronics.

In this session we'd like to explore some of the application and possibilities to be had from the application of advanced microstructural & thermal modeling techniques to further the development and performance of flexible electronics and even piezoelectric fabric. Not only are we looking at wearing electronics. We're looking at having them power themselves.

NICOLAS MONNIER | 1:00 pm ET

Product & Market Specialist, Stäubli North America

PRODUCT DEMO - Quick Connectors for Two-Phase Fluids

Following the adoption of liquid cooling, two-phase fluids are a growing interest in the engineering community due to many thermal advantages. As a leader in the IT Cooling industry, Stäubli will present the main technical considerations for using quick disconnects with two-phase fluids. This product demonstration will cover technical aspects of materials, pressure, tightness, and the Stäubli products designed for these applications.

METODI ZLATINOV[1] & DENVER SCHAFFARZICK[2] | 1:45 pm ET

Technical Lead, R&D[1],Director of Engineering[2], ERG Aerospace

Metal Foams Make Better Heat Exchangers

When faced with high heat fluxes and challenging packaging constraints, it is time to look beyond conventional fins and consider high-performance metal foam heat sinks, cold plates, and heat exchangers. Due to their high surface area, turbulence-enhancing pore structure, and excellent thermal conductivity, open-celled aluminum and copper foams can deliver unrivaled enhancement in heat transfer. With a range of available base materials, pore sizes, porosities, and the freedom to machine, compress, braze, and bond, it is possible to tailor both performance and form factor to the application at hand. In this webinar, ERG Aerospace will dive into the characteristics and properties of metal foams, introduce the principles behind designing effective metal foam cooling systems, and highlight some real-world applications of this technology.

WEDNESDAY | OCTOBER 21, 2020 | *CONTINUED*

JOSEPH FORBES PETRI | 2:45 pm ET

Electronic Assembly Solutions Account Manager, 3M

Thermal Management for a Connected World

In the world of IoT, and an increasing number of connected devices, electronic devices and components are expected to provide greater functionality to meet higher performance requirements. While the operating frequency increases, the amount of power dissipates as heat increases. The buildup of excess heat in electronics can cause 55% of electronic systems to fail. Design engineers are faced with a serious problem – how do we maintain performance and have a more reliable device? The need for thermal management is across multiple market segments. This webinar will focus on thermal management solutions in two key market segments; automotive and aerospace & defense. Join this webinar to learn how 3M™ Thermal Interface Materials can help keep your devices performing at their best.

JOHN WILSON | 3:45 pm ET

Electronics Product Specialist, Siemens Digital Industries Software

Thermal Analysis, Electronics Reliability and the Influences of Variability

The traditional use of electronics thermal analysis in its basic function has always been to assess whether a design will meet requirements representing a worst-case scenario. Improving a design to meet a performance goal using parametric studies, or optimization techniques is widely adopted. Even assessing designs against product use mission profiles using transient simulation is now more widely used whether for power management and efficiency studies or for generating temperature cycling and amplitude insights for lifetime prediction and reliability purposes.

How many thermal simulation-based design processes today encompass the wide influence of variability? By combining electronics cooling CFD simulation with statistical approaches, engineers can more fully explore the design space, verify decisions made, and also evaluate matters such as manufacturing quality variation. This presentation introduces an approach to simulation-based reliability assessment to reduce design process aspects that impact field failure. Studies performed utilized Design of Experiments and Monte Carlo methods and tools, including Simcenter Flotherm software and Simcenter HEEDS design exploration software.

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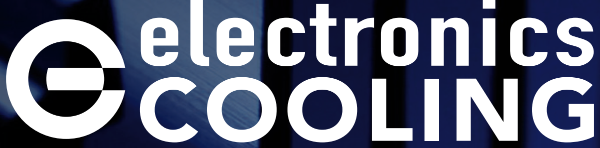

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Advances In High-Performance Cooling For Electronics

Reprinted from the 10th Anniversary Issue, November 2005

Clemens J.M. Lasance
*Philips Research Laboratory

Robert E. Simons
*IBM Corp.

**Note:* *Affiliation as cited in the original article*

EDITOR'S COMMENTS:
Dear Readers, it is my distinct pleasure to share with you this milestone article authored by two former Editors of *Electronics Cooling*. Both Clemens Lasance and Bob Simons had very distinguished careers in their professional lives and during their years as Associate Technical Editors at *Electronics Cooling*. Clemens worked for Philips Research in Eindhoven. Bob served in the IBM Advanced Thermal Laboratory in Poughkeepsie, NY. Both were world-class technical organizations. Over their long careers, they encountered nearly every type of cooling technology allowed by the laws of physics. In this article, originally published in the 10th Anniversary issue of *Electronics Cooling* in November 2005, they described the physical mechanisms operative in 18 classes of cooling methods. Furthermore, they quantify the cooling efficiency of each and evaluate the possibility of further performance improvement in each of them. As one reads this very clear and concisely written article, one can't help but think of the dramatic advances made in many of these fields over the last 15 years. However, it still remains a very useful reference since it catalogs the range of possible cooling options and provides references that can be a starting point for one wishing to get a more in-depth understanding of any of them. In my opinion, because of its encyclopedic breadth, this article represents the high water mark for quality in the 25-year history of *Electronics Cooling*. ***(Bruce Guenin)***

INTRODUCTION

The need for new cooling techniques is driven by the continuing increases in power dissipation of electronic parts and systems. In many instances, standard techniques cannot achieve the required cooling performance due to physical limitations in heat transfer capabilities. These limitations are principally related to the limited thermal conductivity of air for convection and copper for conduction.

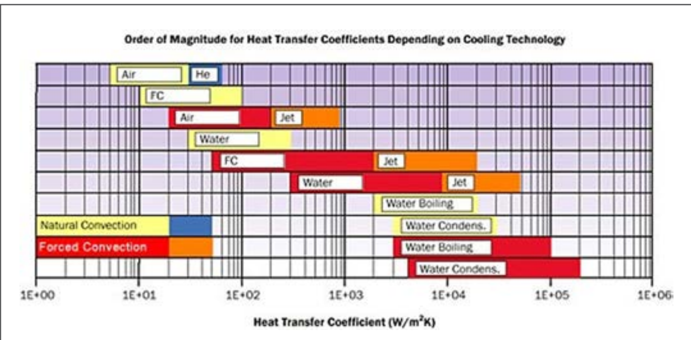


Figure 1: Heat transfer coefficient attainable with natural convection, single-phase liquid forced convection and boiling for different coolants [1].

Figure 1 shows a comparison of various cooling techniques as a function of the attainable heat transfer in terms of the heat transfer coefficient. To accommodate a heat flux of 100 W/cm^2 at a temperature difference of 50 K requires an effective heat transfer coefficient (including a possible area enlarging factor) of 20,000 W/m^{2}K. From *Figure 1* it can be concluded that there will be a need for liquid cooling in the future of thermal management. This article briefly discusses a number of promising thermal management technologies that are emerging for possible electronics applications.

CONDUCTION AND HEAT SPREADING

In all cooling applications, heat from the device heat sources, must first travel via thermal conduction to the surfaces exposed to the cooling fluid before it can be rejected to the coolant. For example, as shown in *Figure 2*, heat must be conducted from the chip to the lid to the heat sink before it can be rejected to the flowing air. As can be seen thermal interface materials (TIMs) may be used to facilitate thermal conduction from the chip to the lid and from the lid to the heat sink. In many cases heat spreaders in the form of a flat plate with good thermal conductivity may be placed between the chip and lid to facilitate spreading of the heat from the chip to the lid or heat sink. Vapor chambers are also used to spread heat from a concentrated chip or module heat source to a larger heat sink.

For high-power applications, the interface thermal resistance becomes an important issue. Direct soldering (e.g., reflow soldering) is often difficult, certainly when copper is used because of the large CTE mismatch between Cu and Si. However, a few promising materials are entering the market.

Diamond-filled greases have been tested to have an effective ther-

mal conductivity of over 20 W/mK; however, the vendor claimed 60 W/mK [2]. Even more interesting is a nanostructured foil, which utilizes a very fast exothermic reaction to create a soldered connection virtually at room temperature [3]. Extensive long-term reliability studies are in progress [4].

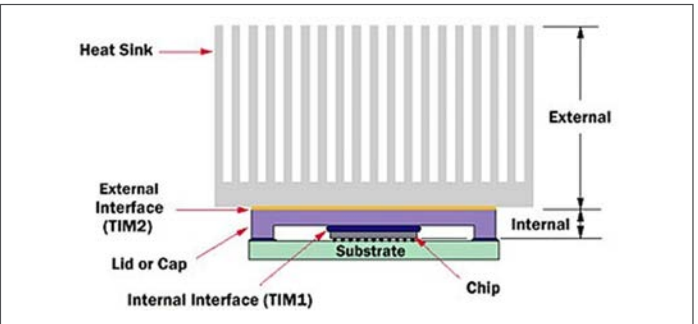


Figure 2: Chip package with thermal conduction path to heat sink via TIMs.

Heat spreading is a very effective way of mitigating the need for sophisticated high-heat-flux cooling options. Of course, to be effective, the benefits of decreasing the heat flux density by increasing the area should outweigh the penalty of adding another layer that the heat must be conducted across. This is an optimization problem as discussed below. The options for advanced heat spreading solutions are two-fold:

- Novel materials such as carbonaceous materials, metal-matrix composites, ceramic matrix composites (e.g., diamond-particle-reinforced silicon carbide), or ScD (Skeleton cemented Diamond), all of them with much higher thermal conductivities than copper, much less weight and tunable CTEs [5].
- Novel heat spreader technologies such as Novel Concept's Isoskin [6] and Enerdyne's Polara [55] that claim effective thermal conductivities that compete with diamond.

By applying heat spreaders cooling methods such as loop heat pipes and low-flow liquid cooling may be augmented to accommodate higher heat flux applications. *Figure 3* provides a graph showing heat spreading results for a 300 W heat source of 2 cm^2 area as a function of thermal conductivity, thickness, and cooling boundary condition (i.e., heat transfer coefficient). Looking at the results it becomes obvious that heat spreading is a complex phenomenon. This is because the conduction and convection effects cannot be separated and the two effects compete: increasing the thickness increases the through-plane resistance but decreases the in-plane thermal resistance. For example, comparing the two upper curves with the two lower curves, their order is changed. The results also show that it is very well possible to use heat spreaders to decrease the required fluid-side heat transfer coefficient to easily manageable values, below 5,000 W/m^{2}K, which could be fairly easily realized with hydrofluoroether (HFE) cooling fluids. For example, using an 8 x 8 cm^2 heat spreader of some advanced composite with a k of 800 W/mK and a thickness of 4 mm results in a temperature rise of 40°C with a heat transfer coefficient of only 2,500 W/m^{2}K.

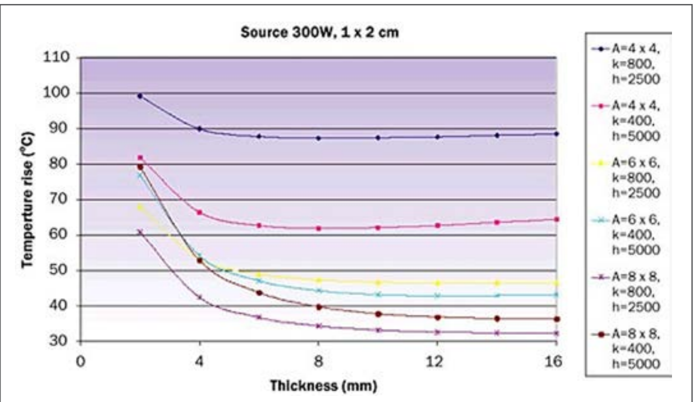


Figure 3: Example of effect of thickness on heat spreading for various heat source areas, material thermal conductivities, and heat transfer coefficients (A in cm^2, k in W/mK, h in W/m^{2}K).

AIR COOLING

It is generally acknowledged that traditional air-cooling techniques are about to reach their limit for cooling of high-power applications. With standard fans a maximum heat transfer coefficient of maybe 150 W/m^{2}K can be reached with acceptable noise levels, which is about 1 W/cm^2 for a 60°C temperature difference. Using 'macrojet' impingement, theoretically we may reach 900 W/m^{2}K, but with unacceptable noise levels. Non-standard fans/dedicated heat sink combinations for CPU cooling are expected to have a maximum of about 50 W/cm^2, which is a factor of 10 higher than expected 15 years ago. However, some new initiatives have emerged to extend the useful range of air-cooling such as piezo fans, 'synthetic' jet cooling and 'nanolightning'.

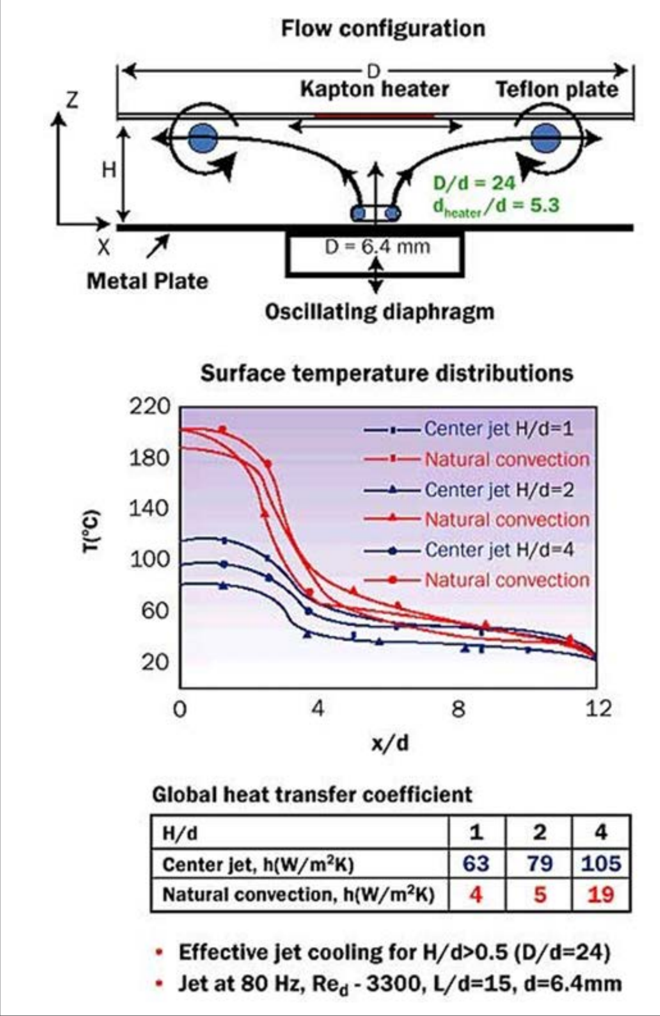
PIEZO FANS

Piezoelectric fans are low power, small, relatively low noise, solid-state devices that recently emerged as viable thermal management solutions for a variety of portable electronics applications including laptop computers and cellular phones. Piezoelectric fans utilize piezoceramic patches bonded onto thin, low frequency flexible blades to drive the fan at its resonance frequency. The resonating low frequency blade creates a streaming airflow directed at electronics components. A group at Purdue reports up to a 100% enhancement over natural convection heat transfer [7].

'SYNTHETIC' JET COOLING

An approach using periodic microjets coined 'synthetic jets' has initially been studied by Georgia Institute of Technology and is being commercialized by Innovative Fluidics. Due to the pulsating nature of the flow, synthetic jets introduce a stronger entrainment than conventional-steady jets of the same Reynolds number and more vigorous mixing between the wall boundary layers and the rest of the flow. One of the test set-ups is shown in *Figure 4*. A synthetic jet entrains cool air from ambient, impinges on the top hot surface and circulates the heated air back to the ambient through the edges of the plate. A radial counter-current flow is created in the gap between the plates with hot air dispersed along the top and ambient air entrained along the bottom surface. The

idea was further explored by the development of flow actuators using MEMS technology [8].



H/d	1	2	4
Center jet, h(W/m²K)	63	79	105
Natural convection, h(W/m²K)	4	5	19

Figure 4: Flow dynamics of normal jet impingement with an oscillating diaphragm.

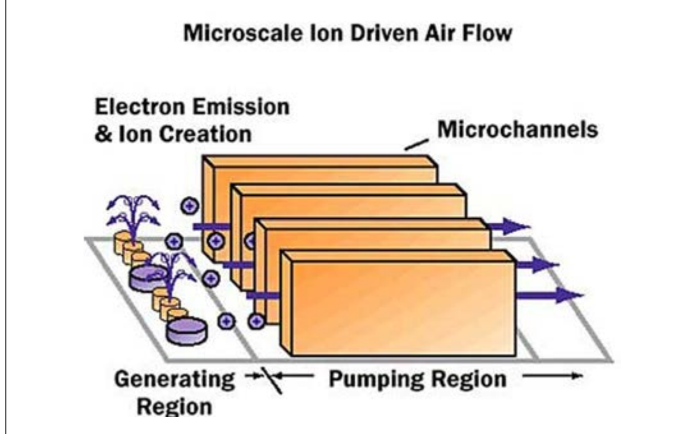


Figure 5: 'Nanolightning' sketch.

'NANOLIGHTNING'

An interesting new approach to increasing the heat transfer coefficient called 'nanolightning' is being pursued by researchers from Purdue. It is based on 'micro-scale ion-driven airflow using very high electric fields created by nanotubes. As shown in *Figure 5*, the ionized air molecules are moved by another electric field, thereby inducing secondary airflow [9]. Cooling a heat flux level of 40 W/cm^2 has been reported. The technology is being commercialized through a start-up company (Thorrn).

LIQUID COOLING

The widely known heat transfer guru John Lienhard [10] once raised the question: "How much heat could possibly be carried away by boiling?" The answer is: 2,000 kW/cm^2 (based on water molecules turning into vapor without influencing each other). The highest reported experimental value is over 200 kW/cm^2, using high velocities and high pressures. Some commercially available microcoolers can handle about 1 kW/cm^2 so there is some room for improvement. Liquid cooling for application to electronics is generally divided into the two main categories of indirect and direct liquid cooling. Indirect liquid cooling is one in which the liquid does not directly contact the components to be cooled. Direct liquid cooling brings the liquid coolant into direct contact with the components to be cooled. The following sections discuss the categories of indirect liquid cooling in the form of heat pipes and cold plates and direct liquid cooling in the form of immersion cooling and jet impingement.

HEAT PIPES

Heat pipes provide an indirect and passive means of applying liquid cooling. They are sealed and vacuum pumped vessels that are partially filled with a liquid. The internal walls of the pipes are lined with a porous medium (the wick) that acts as a passive capillary pump. When heat is applied to one side of the pipe the liquid starts evaporating. A pressure gradient exists causing the vapor to flow to the cooler regions. The vapor condenses at the cooler regions and is transported back by the wick structure, thereby closing the loop. Heat pipes provide an enhanced means of transporting heat (e.g., under many circumstances much better than copper) from a source to a heat sink where it can be rejected to the cooling medium by natural or forced convection. The effective thermal conductivity of a heat pipe can range from 50,000 to 200,000 W/mK [11], but is often much lower in practice due to additional interface resistances. The performance of heat pipes scales from 10 W/cm^2 to over 300 W/cm^2. A simple water-copper heat pipe will on average have a heat transfer capacity of 100 W/cm^2. An example of a typical application of a heat pipe for electronics cooling is given in *Figure 6*.

Although there is virtually no limit to the size of a heat pipe, the effectiveness of a heat pipe decreases with decreasing lengths. For heat pipes with a length less than about 1 cm the performance of a solid piece of metal (e.g., copper) is often comparable. They are very effective as efficient heat conductors to transport heat to locations were more area is available. 2D heat spreaders (otherwise known as vapor chambers) based on the heat pipe principle can achieve much higher effective thermal conductivities than copper.

For example, a thin planar heat spreader has been developed that is claimed to have a thermal performance greater than diamond [12].

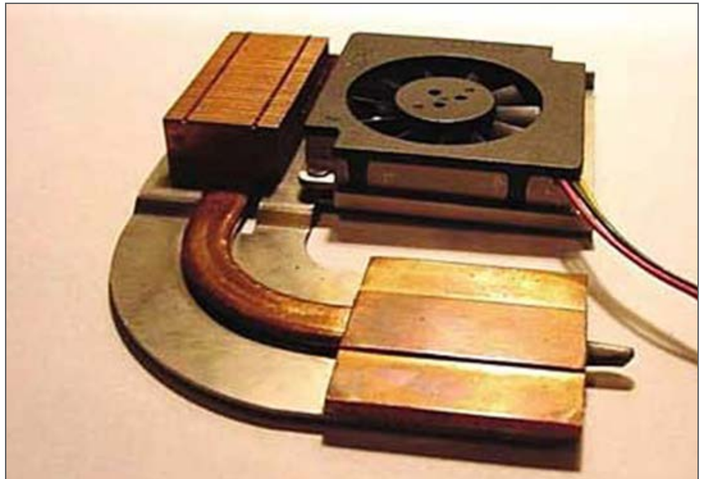


Figure 6: Examples of heat pipes used in a notebook application.

Besides standard heat pipes, loop heat pipes (LHP) such as those shown in *Figure 7* are attracting increased attention. LHPs have the advantage over conventional heat pipes that the vapor and liquid paths are separated enabling much better performance of the liquid return loop. For example, Kim et al. [12] showed the ability to accommodate a heat flux of 625 W/cm^2.

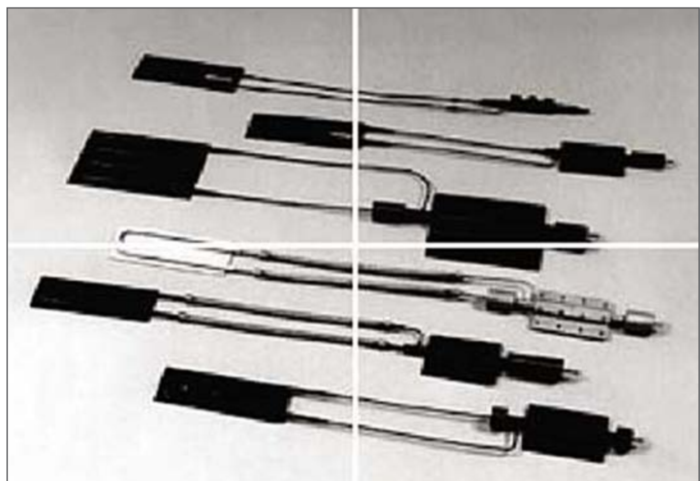


Figure 7: Examples of loop heat pipes.

COLD PLATES

Liquid-cooled cold plates perform a function analogous to air-cooled heat sinks by providing an effective means to transfer heat from a component to a liquid coolant. Unlike heat pipes they may be considered active devices in that liquid is usually forced through them by the action of a pump. For many years water-cooled cold plates were used in mainframe computers to cool high-powered multi-chip processor modules. Vacuum-brazed finstock coldplates are standard practice in defense electronics, and copper-based superalloy structures are used in high-energy lasers. In 1981, in an effort to significantly extend cooling capability, Tuckerman and Pease [13] demonstrated a liquid-cooled microchannel heat sink that removed 790 W/cm^2 with a temperature increase of 71°C for a 600 ml/min flow rate with a pressure drop of 207 kPa. As a result of the continuing increases in heat flux at the chip level, microchannel cold plates are receiving renewed attention.

MICROCHANNELS AND MINICHANNELS

The term 'micro' is applied to devices having hydraulic diameters of 10 to several hundred micrometers, while 'mini' refers to diameters on the order of one to a few millimeters. In many practical cases, the small flow rate within micro-channels produces laminar flow resulting in a heat transfer coefficient inversely proportional to the hydraulic diameter. In other words, the smaller the channel, the higher the heat transfer coefficient. Unfortunately, the pressure drop increases with the inverse of the second power of the channel width, keeping the mass flow constant, and limiting ongoing miniaturization in practice.

Garimella and Sobhan [14] published a very good review of the microchannel literature up to the year, 2000. They concluded, among others, that "Given the diversity in the results in the literature, a reliable prediction of the heat transfer rates and pressure drops in microchannels is not currently possible for design applications such as microchannel heat sinks." Mudawar [15] reviewed high-heat-flux thermal management schemes, including ultra-high heat fluxes in the range of 1,000-100,000 W/cm^2. A recent overview was also provided by Mohapatra and Loikitis [16].

Lee and Vafai [17] compared jet impingement and microchannel cooling for high heat flux applications. One of their conclusions is that microchannel cooling is more effective for areas smaller than 7 x 7 cm. Integrated single and two-phase micro heat sinks are treated by Gillot et al. [18]. They were able to cool about 450 W/cm^2 using both single- and two-phase heat transfer. For two-phase flow the pumping power is about ten times lower and the required flow rate is considerably lower. Kandlikar and Upadhye [19] showed enhanced microchannel cooling by using off-set strip fins and a split-flow arrangement. Cooling of over 300 W/cm^2 at 24 kPa is claimed with a flow of 1.5 l/min. A paper devoted to pumping requirements has been written by Singhal et al. [20]. Useful graphs compare the performance of a whole range of pumps that could be considered for microchannel cooling. Colgan et al. [21] at IBM published a practical implementation of a silicon microchannel cooler (as shown in *Figure 8*) for high-power chips.

They argued that it is not practical to form the microchannels directly on the chip given the high cost of high-performance processor chips. Instead, a separate microchannel cold plate is bonded to the back of the chip. This requires a very low interface thermal resistance. If the microcooler is based on silicon, a rigid bonding means that silver-filled epoxies or solder should be used. Power densities in excess of 400 W/cm^2 are reported, for a flow of 1.2 l/min at 30 kPa.

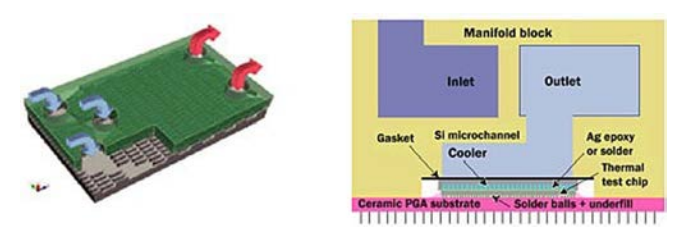


Figure 8: Pictures from IBM paper showing high-performance liquid cooling technology using microchannels [21].

It may be possible to push microchannel heat transfer even further by utilizing boiling. In addition to offering higher heat transfer coefficients, boiling convection in microchannels is promising because it requires less pumping power than single-phase liquid convection to achieve a given heat sink thermal resistance. For the same heat flux the pressure drops by a factor of 20. A review has been published by Bergles et al. [22]. A prototype of a 1,000 W/cm^2 cooling system based on boiling heat transfer in microchannel heat sinks using a flow rate of 500 ml/min has been described in [23]. The main practical problem with two-phase flow is its unpredictability. Local heat transfer coefficients may change appreciably over time leading to local temperature changes of 10-15°C [24]. Also backflow of already heated flow due to expansion of bubbles is observed.

Increasingly, microchannel-like cold plates are becoming commercially available. Mikros [25] claims 1,000 W/cm^{2}K, 14-21 kPa and 0.05 K/W/cm^2 for its patented technology in which the fluid impinges on the surface to be cooled. ACT (Advanced Cooling Technologies) [26] offers pumped liquid (both single- and two-phase) cooling technologies in addition to loop heat pipes for space applications. Their single-phase solution (see *Figure 9*) incorporates a 'unique oscillating flow heat transfer' mechanism, capable of cooling over 1,300 W/cm^2.

Another company, Cirrus, sells microchannel high heat flux heat sinks, claiming over 100 W/cm^2 heat flux capability and reduced pressure drops. Recently another player, iCurie [27], entered the market. They claim a pumpless microchannel cooling system based on capillary pumping with separate liquid-vapor loops. The layout can be very flexible, only 300 micrometer thick. At least 200 W/cm^2 can be cooled, with the additional advantage of a negligible effect of gravity. It is not a heat pipe because the principle of operation is based on flow boiling, not evaporation. Other companies offering microchannel coolers include Lytron, Novel Concepts, Curamik, Microcooling Concepts (MC^2), and Koolance.

A completely different way of making microchannels is by using metal foams or metal made porous otherwise. Engineers from Thermacore have described a family of liquid-cooled heat sinks using porous metal. They accommodated a heat flux of 500 W/cm^2 for a 50 K difference at a pressure drop of 115 kPa using water [28].

On June 21, 2005, Georgia Tech [29] announced a novel monolithic technique for fabricating liquid cooling channels onto the backs of high-performance ICs. They also built a system that would allow the on-chip cooling system to be connected to embedded fluidic channels built into a printed circuit board.

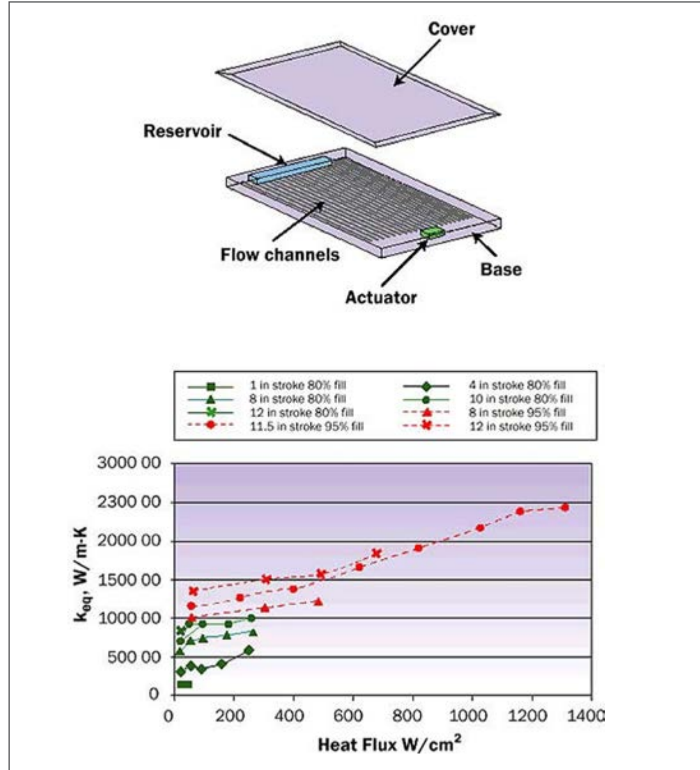


Figure 9: Vendor's heat spreader and test results.

ELECTROHYDRODYNAMIC AND ELECTROWETTING COOLING

As an alternative to a continuous flow set into motion by either temperature differences or by mechanical means, liquid could also be formed and moved in droplets of nano-to-milliliter size (see for example a nice demonstration by Nanolytics [30]) by means of electric fields.

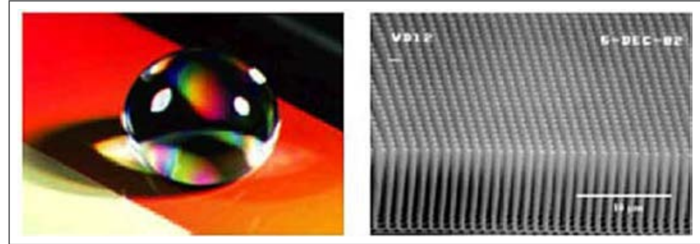


Figure 10: Water forms a nearly perfect ball, as shown in left photo, suspended on the tips of tiny blades of nanograss.

Electrowetting on a dielectric film, in which the surface property of a dielectric film can be modified between hydrophobic and hydrophilic states using an electric field, can be used to provide the basis for a direct micropumping system. Electrowetting involves control of the surface tension of a liquid and can cause a droplet of

liquid to bead (as shown in *Figure 10*) or spread out on the surface depending upon its surface state.

One of the possible applications is cooling on a micro scale. The recently published theoretical work of Pamula [31] has shown a possible configuration based on fast moving droplets under a chip. They showed that with 0.4 ml/min it is theoretically possible to cool 90 W/cm². Recently, Leuven University in collaboration with Philips Research published two papers on this subject [32, 33]. The Philips approach differs from the Duke approach in that it concerns an oscillating flow. At Bell Labs researchers coupled electrowetting with nanostructured superhydrophobic surfaces (coined 'nanograss') to result in dynamically tunable surfaces [34]. One application is the movement of droplets to cool hot spots; however, no further heat transfer data are given.

A recent publication discussed another promising development, the application of electrowetting to liquid metals [35]. The main advantage besides a better heat transfer capability is the much lower voltage required (2 instead of 50 V). However, no experimental data have been presented.

The interesting aspect in combining microfluidics with electric control is that when all sizes scale down to micro scale, the electro/-kinetic/-wetting/-osmotic forces become comparable to pressure drop forces and therefore control of the liquid motion becomes easier. Of course, active cooling of a hot surface is one thing, to remove heat from the heated liquid in a closed loop requires additional heat exchange area.

LIQUID METAL COOLING

Of special interest is the work ongoing in the field of liquid metal cooling. Apart from heat pipes based on liquid metals, mainly for the high-temperature range, an increasing amount of research is devoted to the use of Ga-Sn-In eutectics that remain liquid down to -19°C. In [36, 37] high-performance liquid metal cooling loops are described using magneto fluid dynamic pumps, claiming over 200 W/cm² cooling capacity, using a flow of 0.3 l/min at 15 kPa. Examples of liquid cooling loops for electronics cooling application are shown in *Figures 11* and *12*. Another advantage of liquid metal is its much lower CTE compared to water and the fact that freezing introduces fewer problems. Developments to extend the use in cold environments to -40°C are ongoing.

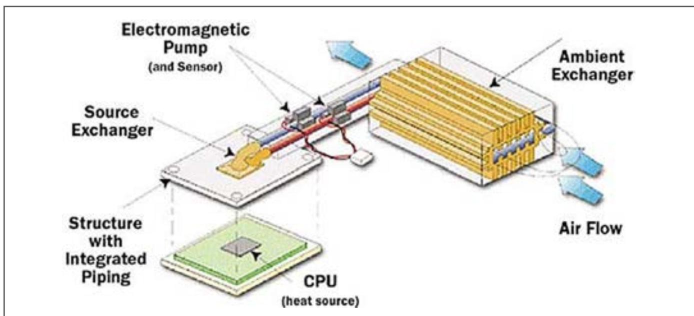


Figure 11: Sketch of liquid metal cooling loop.

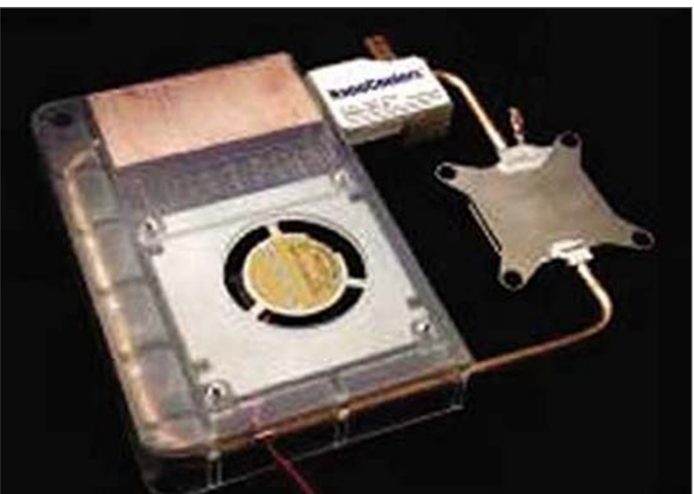


Figure 12: Typical configuration of liquid metal cooling loop for mobile applications.

IMMERSION COOLING

Direct liquid or immersion cooling is a well-established method for accommodating high heat flux backed by over 30 years of university and industrial research. With natural convection two-phase flow, generally termed nucleate pool boiling, the critical heat flux using FC-72 is in the range of 5 to 20 W/cm². However, much higher heat fluxes up to 100 W/cm² can be accommodated with surface enhancement of the heat source. *Figure 13* illustrates a device submerged in a pool of dielectric liquid. The heat dissipated in the device produces vapor bubbles that are driven by buoyancy forces into the upper region of the container, where the vapor condenses and drips back into the liquid pool. One of the disadvantages of this technique is the need for a liquid compatible with the device. Most often, water cannot be used because of its chemical and electrical characteristics.

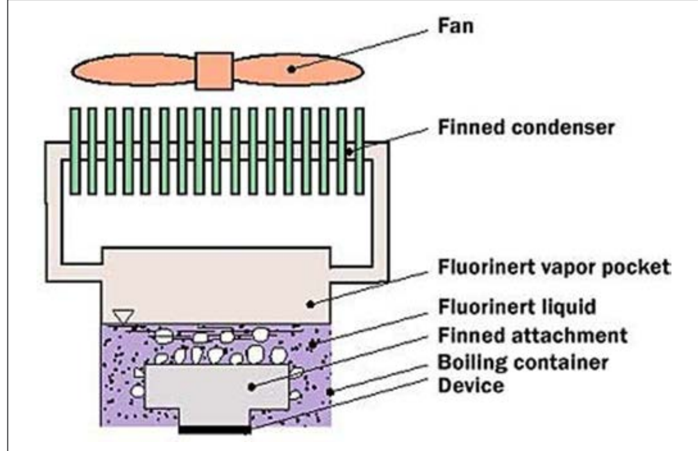


Figure 13: Example of pool boiling (thermosyphon) cooling.

LIQUID JET IMPINGEMENT

Wang et al. [38] claim a cooling of 90 W/cm² with a 100°C temperature rise using a flow rate of only 8 ml/min. Researchers at Georgia Institute of Technology studied a closed loop impingement jet [39]. Cooling of almost 180 W/cm² has been realized us-

ing water, using a flow of 0.3 l/min at 300 kPa. The micropump used 7 W to drive it.

At this point it is difficult to say which one is better, microchannels or microjets. Microchannels are easier to fabricate and implement but the temperature nonuniformity is larger and the nucleation is more difficult to control. Microjets achieve better cooling uniformity but more fabrication steps are required and an initial pressure is necessary to form the jet. An example of a commercial concept for impingement liquid cooling is shown in *Figure 14* [40].

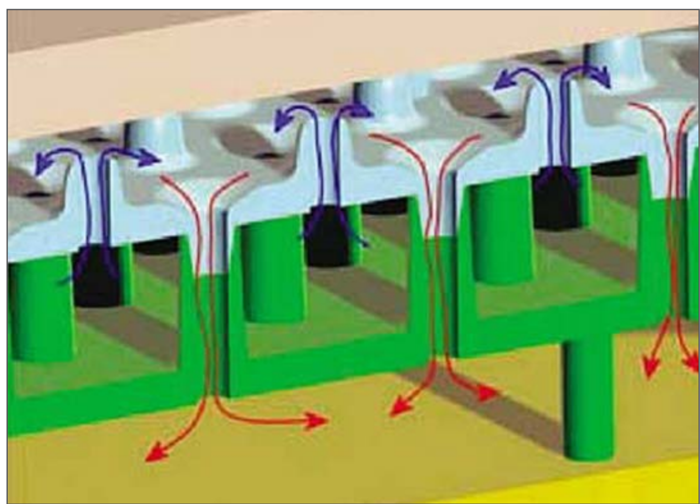


Figure 14: Commercially available multiple jet impingement liquid cooling.

SPRAY COOLING

In recent years spray cooling has received increasing attention as a means of supporting higher heat flux in electronic cooling applications. Spray cooling breaks up the liquid into fine droplets that impinge individually on the heated wall. Cooling of the surface is achieved through a combination of thermal conduction through the liquid in contact with the surface and evaporation at the liquid-vapor interface. The droplet impingement both enhances the spatial uniformity of heat removal and delays the liquid separation on the wall during vigorous boiling.

Spray evaporative cooling with a Fluorinert™ coolant is used to maintain junction temperatures of ASICs on MCMs in the CRAY SV2 system between 70 and 85°C for heat fluxes from 15 to 55 W/cm^2 [41]. In addition to the CRAY cooling application, spray cooling has gained a foothold in the military sector providing for improved thermal management, dense system packaging, and reduced weight [42]. A research group at UCLA discussed chip-level spray cooling for an RF power amplifier and measured a maximum heat flux of over 160 W/cm^2 [43]. Isothermal Systems Research manufactures SprayCool products [44].

Spray cooling and jet impingement (as shown in *Figure 15*) are often considered competing options for electronic cooling. In general, sprays reduce flow rate requirements but require a higher nozzle pressure drop.

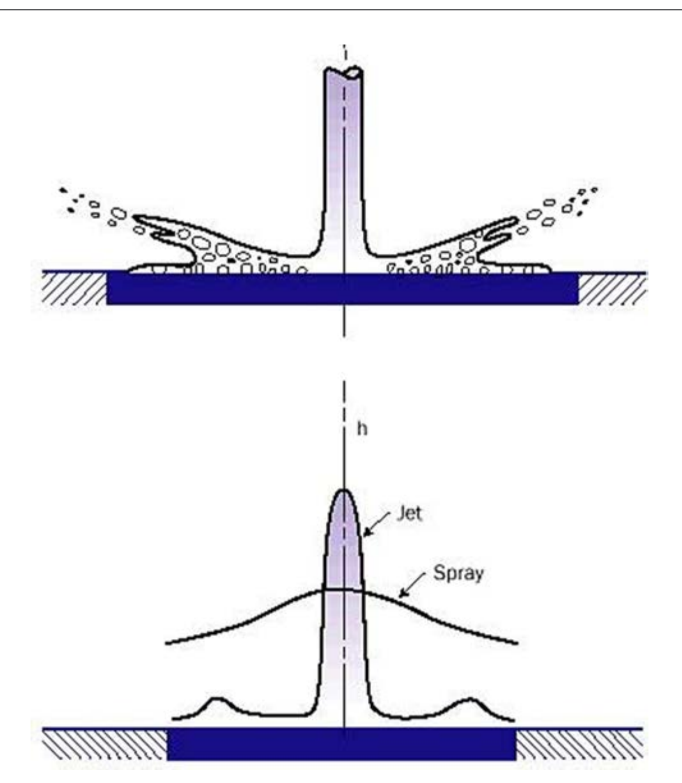


Figure 15: Illustration of spray and jet impingement cooling.

A final method to be mentioned is inkjet-assisted spray cooling. This method uses existing thermal inkjet technology. A critical heat flux of 270 W/cm^2 is reported [45] using only 3 ml/min, with a COP of six, meaning that the inkjet pumping power is a factor of six lower than the heat removed.

SOLID-STATE COOLING

A thermoelectric or a Peltier cooler (as shown in *Figure 16*) is a small electronic heat pump that has the advantage of no moving parts and silent operation. Thermoelectric cooling enables cooling below ambient temperature. The coolers operate on direct current and may be used for heating or cooling by reversing the direction of current flow.

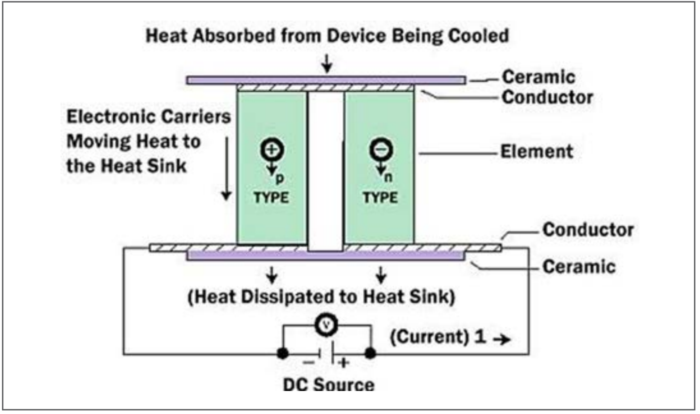


Figure 16: Schematic of simple Peltier cooler.

When a positive DC voltage is applied to the n-type thermoelement, electrons pass from the p- to the n-type thermoelement and the cold side temperature decreases as heat is absorbed.

The heat absorption (cooling) is proportional to the current and the number of thermoelectric couples. The main disadvantage is that the heat transferred to the hot side is greater than the amount of heat pumped by a quantity equal to the Joule heating (i.e.: I^2R loss) that takes place in the Peltier elements.

The three most important thermoelectric effects are the Seebeck, Peltier, and Thomson effects. For thermoelectric cooling the Thomson effect can be neglected. The Peltier coefficient Π and Seebeck coefficient S are related to each other through $\Pi = S \bullet T$. Thermoelectric materials are usually characterized by their Figure of merit ZT, defined by:

$$ZT = \frac{\sigma S^2 T}{\lambda}$$

Where σ = electrical conductivity
λ = thermal conductivity
T = temperature in absolute units

This equation shows why it is difficult to obtain good thermoelectric materials. A good heat conduction between the hot and cold side) and a high electrical conductivity (to minimize Joule heating). Due to the dependence of σ and λ (Wiedemann-Franz law), it is almost impossible to optimize this ratio for the electron contribution to the thermal conductivity. Hence, the approach is to reduce the phonon thermal conductivity without a degradation of the electrical conductivity. The best materials so far are alloys of Bi_2Te_3 with Sb_2Te_3 and Bi_2Te_3 with Bi_2Se_3. ZT is of the order of one at room temperature. This value gives a Coefficient of Performance (COP, a measure for the efficiency) of about 1 (see *Figure 17a*), which compared to household refrigerators and air conditioners (COP from 2 to 4), makes thermoelectric cooling generally not competitive. The same holds for power generation (see *Figure 17b*).

Despite the low efficiency, the application areas are increasing and include infrared detector cooling, charge coupled devices (CCDs), microprocessors, blood analyzers, portable picnic coolers. Principal applications are still accurate control of temperature and cooling below ambient temperature.

One of the problems with traditional Peltier elements is their limited capability of cooling heat fluxes over 5-10 W/cm². Because the cooling density of a Peltier cooler is inversely proportional to its length, scaling to smaller size is desirable. The material structure produced by conventional crystal growth techniques for producing bismuth telluride thermoelectric materials impose significant limitations on thermoelectric element dimensions due to poor manufacturing yields. This prevents thermoelectric elements from being made very short. Marlow Industries reported new fine-grain micro-alloyed bismuth telluride materials that do not suffer the element geometry limitation and can offer higher performance [47]. Another serious step forward has been realized by Nanocoolers through a proprietary wafer-scale manufacturing process. It concerns a monolithic process with thicknesses about 1-2 micrometers (see *Figure 18*). They claim a tunable performance of 10-1,000 W/cm² with a single stage ΔT of 50-70 K [48]. Micro-Pelt, a spin-off company from Fraunhofer and Infineon also sells promising thin film thermoelectrics (see *Figure 19*). For their near-future products they claim cooling of 160 W/cm².

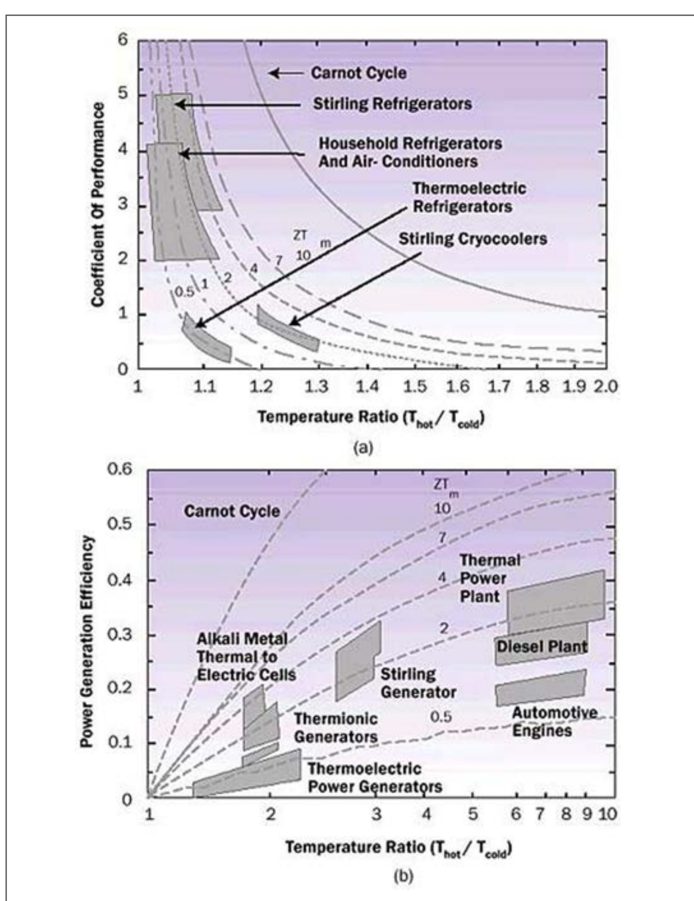


Figure 17: Comparison of thermoelectric technology with other energy conversion methods for (a) cooling and (b) power generation [46].

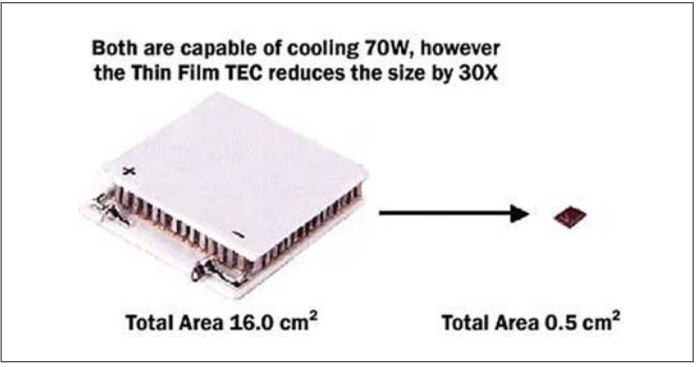


Figure 18: Comparison of standard and thin-film Peltier element.

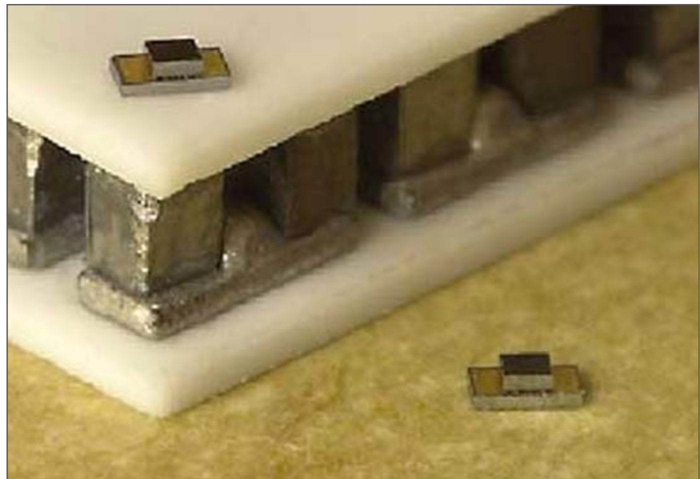


Figure 19: Thin film Peltier element.

A number of other research projects directed at miniaturizing Peltier elements are worth mentioning. In 2004, Biophan Technologies [49] signed an agreement with NASA for characterization and joint development of high-density nanoengineered thermoelectric materials for use with implantable devices. They anticipate a breakthrough in power generation systems. A French/U.S. consortium published a paper [50] on the fabrication and modeling of an in-plane thermoelectric micro-generator. They concluded that a heating power of about 100 mW may be enough to produce 1 mW of useful electrical power in vacuum, using thin film technology. A compact thermoelectric device may be able to produce 60 microwatt with an output voltage of 1.5 V in air. Applied Digital Solutions uses its ThermoLife thermoelectric power generator to power its implantable chips [51]. It generates 3V, has a diameter of 9.3 mm and weighs 230 mg. DTS in Germany uses thin film technology for their Low Power Thermoelectric Generators (LPTGs) [52] that produce a few tens of microwatts in the volt range for a temperature difference of a few degrees C. The generators have a mass of 390 mg. Also in Germany, researchers at Dresden University [53] have found a way to make tiny thermoelectric generators using copper foil as a template. Antimony-Bismuth thermocouples are electro-deposited and after adding an epoxy film the copper is etched away. The result is a cheap, flexible and recyclable generator that converts environmental heat into electricity. The growth by pulsed laser deposition of high-quality thermoelectric cobaltate thin films on silicon has been reported by Yu et al. [54]. In addition, TEM characterization revealed nearly perfect crystalline structures of the $Ca_3Co_4O_9$ film formed on top of an SiO_x amorphous layer, suggesting self-assembly might be a viable technique for cobalt oxide-based thermoelectrics.

If the application is limited to temperatures above ambient temperature matters are quite different. Because the heat flow paths of the cooling and the conduction have the same direction, high-thermal conductivity materials are preferred. Enerdyne's Polara heat spreaders are based on this principle [55]. A factor of five improvement compared to copper is claimed as shown in *Figure 20*. However, no samples are available as of July 2005.

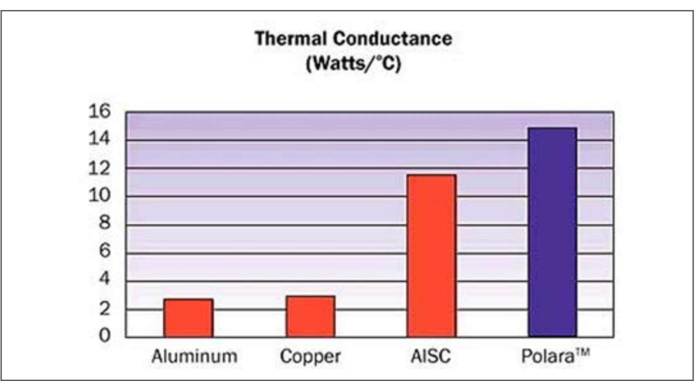


Figure 20: Comparison of effective thermal conductance.

SUPERLATTICE AND HETEROSTRUCTURE COOLING

For a number of years now, the strategy to improve thermoelectric cooling has taken a new turn giving hope for the future. On a nano scale level, coherent and incoherent transport plays an important role in the electron and phonon diffusion. Extensive research is going on in this field. For example, Venkatasubramanian [56] at RTI reported ZT values between two and three at room temperature obtained with Bi_2Te_3/Se_2Te_3 superlattices. Cooling power density is estimated as high as 700 W/cm^2 at 353 K compared to 1.9 W/cm^2 in the bulk material (see *Figures 21* and *22*).

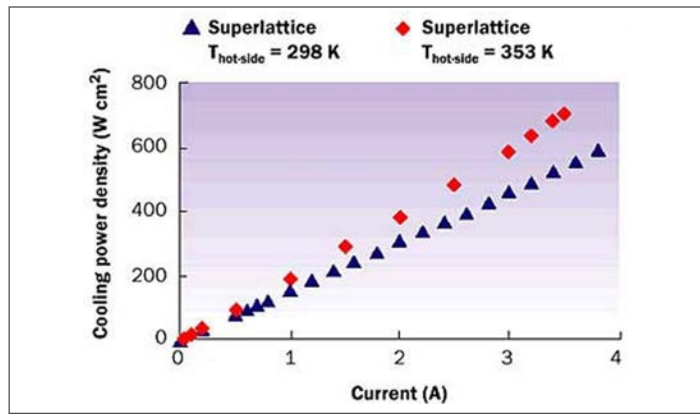


Figure 21: Estimated power density for superlattice devices as a function of current

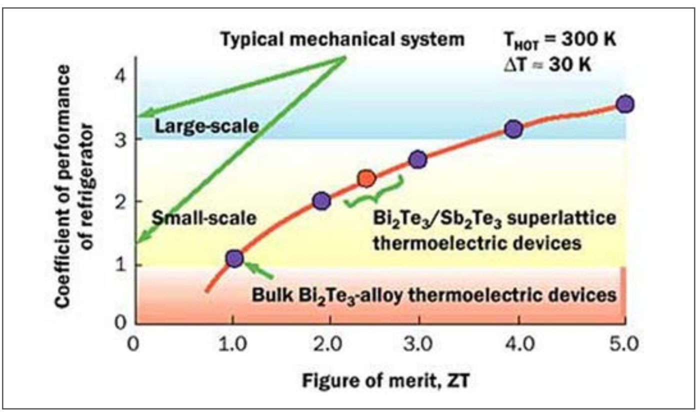


Figure 22: Potential COP as a function of ZT with various technologies. T_{HOT} refers to the heat sink temperature

Thin-film related work is also being conducted at the University of California Santa Cruz, based on SiGe/Si. The most recent paper [57] quotes a cooling power density of nearly 600 W/cm^2 for a temperature difference of 4K below ambient for a 40 x 40 micrometer size area. The superlattice efforts of RTI are being commercialized through a spin-off company called Nextreme. Recent information reveals that despite their claimed value of ZT = 2.4, they are not able to manufacture production samples with a ZT larger than 1.4. The focus is to reduce the parasitics and to reduce even further the current 100 micrometer thickness.

However, there is still hope for a serious breakthrough. Very recently, Humphrey and Linke [58] published a paper called "Reversible Thermoelectric Materials." They argue that nanostructured materials with sharply peaked electronic density of states (such as quantum wires) may operate reversibly, challenging the view that thermoelectric devices are inherently irreversible heat engines. In this case, ZT values could reach a value of 10 at room temperature, much above the value of 5 that is required for economical adoption of thermoelectric technology for mainstream refrigeration and power generation.

THERMIONIC AND THERMOTUNNELING COOLING

Thermionic cooling is based on the principle that a high-work-function cathode preferentially emits hot electrons [46]. Materials available have a work function of 0.7 eV or higher, which limits the use to the higher temperature ranges (>500 K). Vacuum thermionic devices based on resonant tunneling have been proposed more recently [59]. Cooling capabilities of 20-30°C with kW/cm^2 cooling power density can be achieved. However, since the operating currents for the device are as high as 105 A/cm^2, effects such as Joule heating at the metal-semiconductor contact resistance and reverse heat conduction have limited the experimental cooling results to <1°C.

Devices based on quantum tunneling through a small gap are being commercialized [60]. The spacing between the cathode and the anode should be of the order of 10 nm, providing quite an engineering challenge. Much larger cooling power than thermoelectric superlattice coolers are predicted by Hishinuma et al. [61] (e.g. 10 kW/cm^2 for 50 K cooling at room temperature).

Recently a study has been devoted to their potential use as energy scavenging or power conversion devices [62]. Unfortunately, the conclusion is that a gap an order of magnitude lower must be achieved to be of interest for these application fields. Even more worrying is another recent study [63] showing that contrary to the results of Hishinuma only about 16 W/cm^2 can be reached with a ΔT of 40°C, while the maximum COP is only 0.25. More or less the same conclusion can be drawn from a paper presented at THERMINIC 2005, September 27-30 [64].

Herein, some weaknesses in prior studies are discussed and it is clear from the conclusions that nanogap solutions without significant improvements in lower work function materials have no future.

PHASE CHANGE MATERIALS AND HEAT ACCUMULATORS

Phase change materials are successfully used as heat-storing materials for air conditioning, cool boxes, efficient fire-retarding powders, as functional materials for self-heating insoles for boots and many other industrial applications. Their use for electronics thermal management is limited to applications where time-dependent phenomena play a role. For example, reference [65] discusses the use of phase change materials as compared to copper for use in a power semiconductor unit.

Chemical heat accumulators should also be mentioned. For example, the use of composite materials based on granulated open-porous matrix filled with a hygroscopic substance can be seen as a new approach to accumulate heat [66]. The advantage is a significant increase in the heat that can be stored as compared to sensible heat and latent heat. For example, for a 100°C temperature rise copper absorbs 40 kJ/kg. Evaporation of water is associated with an absorption of 2,260 kJ/kg. The enthalpy of a reversible chemical reaction can reach a value of 7,000 kJ/kg. A principal advantage of reversible chemical reactions for heat accumulation is their ability to store the accumulated energy for a long time, if the reaction is controlled by the presence of either a catalyst or a reagent. Hence, the major applications are in the field of summer-winter heat storage for buildings, etc. Chemical heat accumulators could potentially be used for outdoor electronic applications when a night-day rhythm is present.

CONCLUSIONS

A number of approaches show interesting industrial potential for the cooling of high-power electronics. This prospect is attested to by the number of small companies that are entering the market. For example, there are now companies engaged in the development and commercialization of microchannels, spray cooling, synthetic jets, thin film Peltier elements. For heat flux densities up to and maybe even beyond 50 W/cm^2 air-cooling may remain the cooling option of choice. For heat fluxes over 100 W/cm^2, some form of liquid-cooling appears to be the most viable option. Several papers have demonstrated solutions that may be industrially feasible for application in the range between 500 and 1,000 W/cm^2. Considering the range of efforts underway to extend conventional cooling technologies, as well as develop new ones, the future seems bright for accommodating high-heat flux applications.

(Note: We deemed it instructive to include examples of commercially-available thermal solutions. However, it should be clearly stated that we do not intend to promote any of the mentioned products. Finally, we have tried to cover the state-of-the-art as known to us. However, given the broadness of the field, we may have overlooked some important new developments for which we apologize.)

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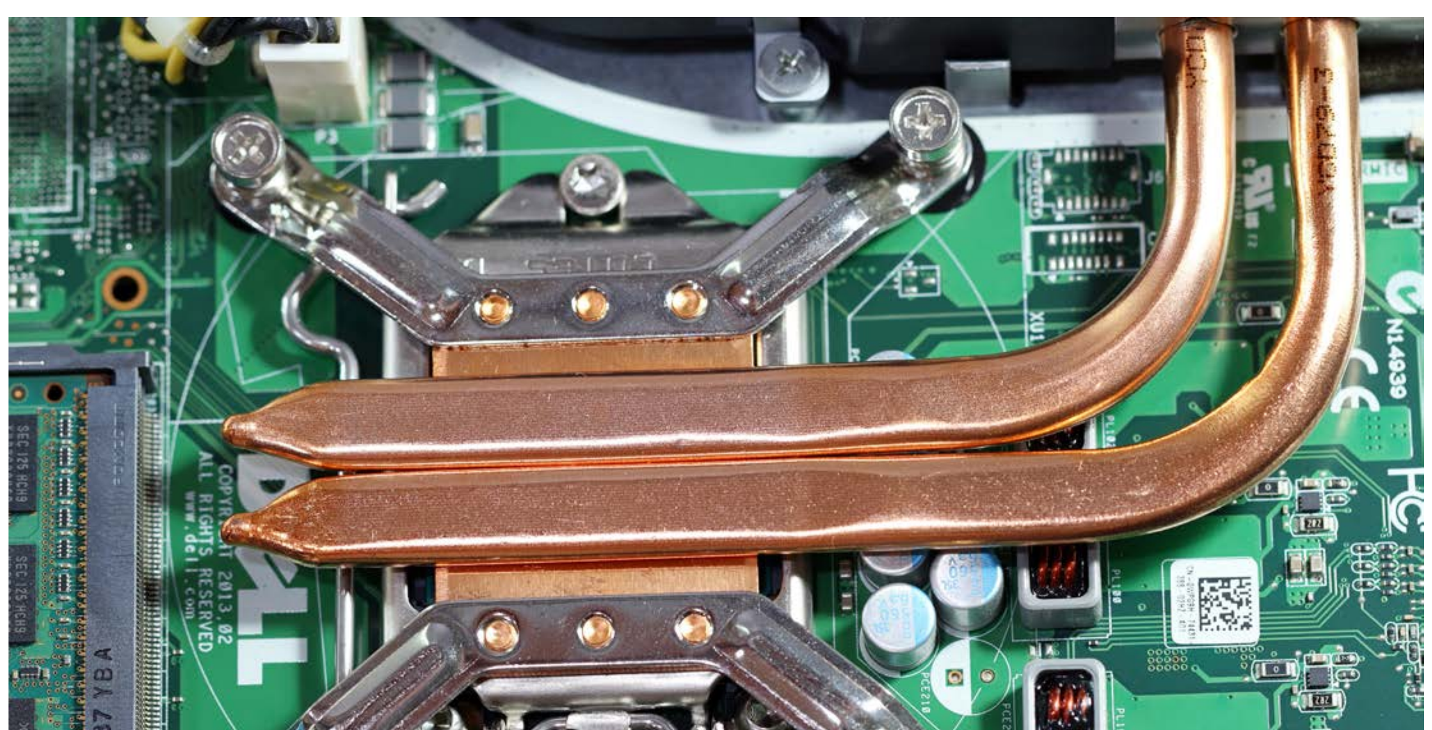

Estimating Parallel Plate-Fin Heat Sink Thermal Resistance

Reprinted from February 2003

Robert E. Simons
*IBM Corp.
Note: *Affiliation as cited in the original article*

EDITOR'S COMMENTS:
Over the last 20 years, I have collected many simple tools into a spreadsheet that I can use for making quick engineering estimates. One prominent worksheet in that collection is a calculator that makes use of this "Calculation Corner" and its companion article, "Estimating Parallel Plate-fin Heat Sink Pressure Drop", to predict the thermal performance of plate fin heat sinks. This is just one example of the many extremely valuable "Calculation Corner" columns that Bob Simons contributed to the entire electronics cooling industry through *Electronics Cooling.* ***(Ross Wilcoxon)***

INTRODUCTION

As noted previously in this column, the trend of increasing electronic module power is making it more and more difficult to cool electronic packages with air. As a result, there are an increasing number of applications that require the use of forced convection air-cooled heat sinks to control module temperature. An example of a widely used type of heat sink is the parallel plate configuration shown in *Figure 1*.

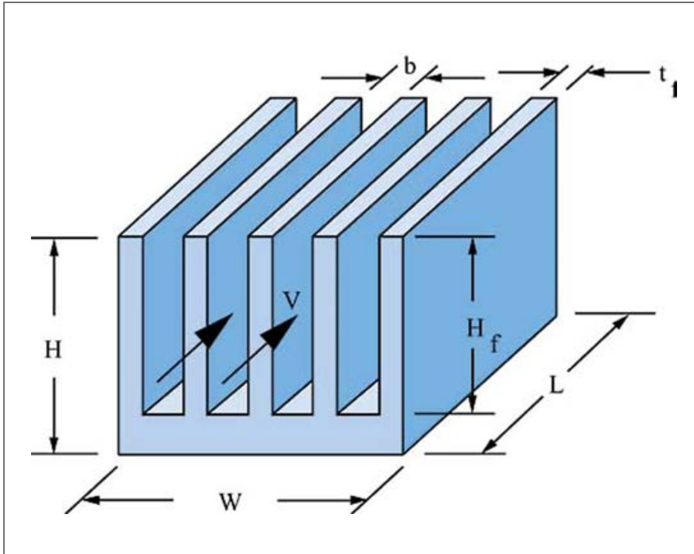

Figure 1: Parallel plate fin heat sink configuration.

In order to select the appropriate heat sink, the thermal designer must first determine the maximum allowable heat sink thermal resistance. To do this it is necessary to know the maximum allowable module case temperature, T_{case}, the module power dissipation, P_{mod}, and the thermal resistance at the module-to-heat sink interface, R_{int}. The maximum allowable temperature at the heat sink attachment surface, T_{base}, is given by

$$T_{base} = T_{case} - P_{mod} * R_{int} \quad (1)$$

The maximum allowable heat sink resistance, R_{max}, is then given by

$$R_{max} = \frac{T_{base} - T_{air,in}}{P_{mod}} \quad (2)$$

where $T_{air,in}$, is the temperature of the cooling air at the inlet to the heat sink passages. At this point many thermal engineers will start looking at heat sink vendor catalogs (or more likely today start searching vendors on the internet) to find a heat sink that will fit in the allowable space and provide a heat sink thermal resistance, R_{hs}, less than R_{max} at some specified flow rate. In some cases, it may be useful to do a sizing to estimate R_{hs} for various plate-fin heat sink designs to determine if a feasible design configuration is possible. The remainder of this article will provide the basic equations to do this. The thermal resistance of the heat sink is given by

$$R_{hs} = \frac{1}{h * (A_{base} + N_{fin} * \eta_{fin} * A_{fin})} \quad (3)$$

where h is the convective heat transfer coefficient, A_{base} is the exposed base surface area between fins, N_{fin} is the number of fins, η_{fin} is the fin efficiency, and A_{fin} is the surface area per fin taking into account both sides of the fin.

To proceed further it is necessary to establish the maximum allowable heat sink volume in terms of width, W, height, H, and length in the flow direction, L. It is also necessary to specify a fin thickness, t_{fin}. Using these parameters, the gap, b, between the fins

may be determined from

$$b = \frac{W * N_{fin} * t_{fin}}{N_{fin} - 1} \quad (4)$$

The exposed base surface area may then be determined from

$$T_{base} = T_{case} - P_{mod} * R_{int} \quad (5)$$

and the heat transfer area per fin from

$$A_{fin} = 2 * H_f * L \quad (6)$$

At this point it is necessary to specify the air flow rate either in terms of the average velocity, V, between the fins or a volumetric flow rate, G. If a volumetric flow rate is used, the corresponding air velocity between the fins is

$$V = \frac{G}{N_{fin} * b * H_f} \quad (7)$$

To determine the heat transfer coefficient acting upon the fins, an equation developed by Teertstra et al. [1] relating Nusselt number, Nu_b, to Reynolds number, Re, and Prandtl number, Pr, may be employed. This equation is

$$Nu_b = \left[\frac{1}{\left(\frac{Re * Pr}{2}\right)^3} + \frac{1}{\left(0.664\sqrt{Re}Pr^{0.33}\sqrt{1+\frac{3.65}{\sqrt{Re}}}\right)^3} \right]^{-0.33} \quad (8)$$

The Prandtl number is

$$Pr = \frac{\mu c_p}{k} \quad (9)$$

where μ is the dynamic viscosity of air, c_p the specific heat of air at constant pressure, and k is the thermal conductivity of air. The Reynolds number used in (8) is a modified channel Reynolds number defined as

$$Re = \left(\frac{\rho V}{\mu}\right) * \left(\frac{b}{L}\right) \quad (10)$$

where ρ is the density of air. *Equation (8)* is based upon a composite model spanning the developing to fully developed laminar flow regimes and was validated by the authors [1] by comparing with numerical simulations over a broad range of the modified channel Reynolds number ($0.26 < Re_b < 175$) and with some experimental data as well.

Using the Nusselt number obtained in (8) the heat transfer coefficient is given by

$$h = Nu_b * \frac{k_{fluid}}{b} \quad (11)$$

where k_{fluid} is the thermal conductivity of the cooling fluid (i.e. air). The efficiency of the fins may be calculated using

$$\eta_{fin} = \frac{\tanh(m * H_f)}{m * H_f} \quad (12)$$

where m is given by

$$m = \sqrt{\frac{2h}{k_{fin} t_{fin}}} \quad (13)$$

and k_{fin} is the thermal conductivity of the fins.

Using these equations, it is possible to estimate heat sink thermal performance in terms of the thermal resistance from the temperature at the base of the fins to the temperature of the air entering the fin passages. It may be noted that the relationship for Nusselt number (8) includes the effect of the temperature rise in the air as it flows through the fin passages. To obtain the total thermal resistance, R_{tot}, to the base of the heat sink it is necessary to add in the thermal conduction resistance across the base of the heat sink. For uniform heat flow into the base R_{tot} is given by

$$R_{total} = R_{hs} + \frac{H - H_f}{k_{base} * W * L} \quad (14)$$

and k_{base} is the thermal conductivity of the heat sink base.

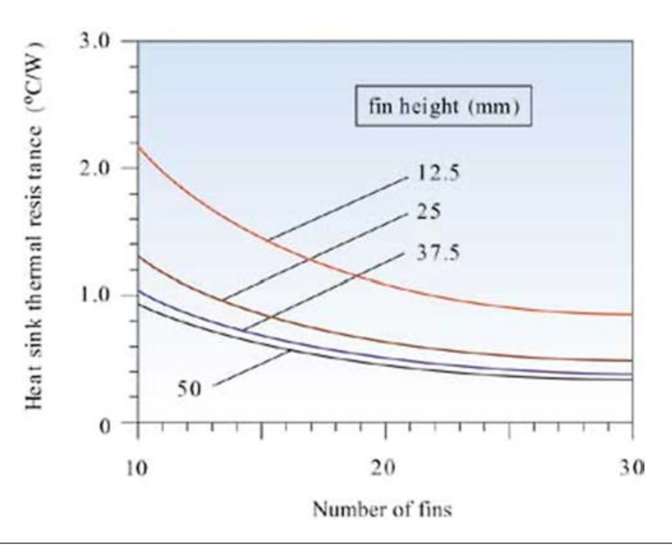

Figure 2: Effect of fin height and number of fins on heat sink thermal resistance at an air velocity of 2.5 m/s (492 fpm).

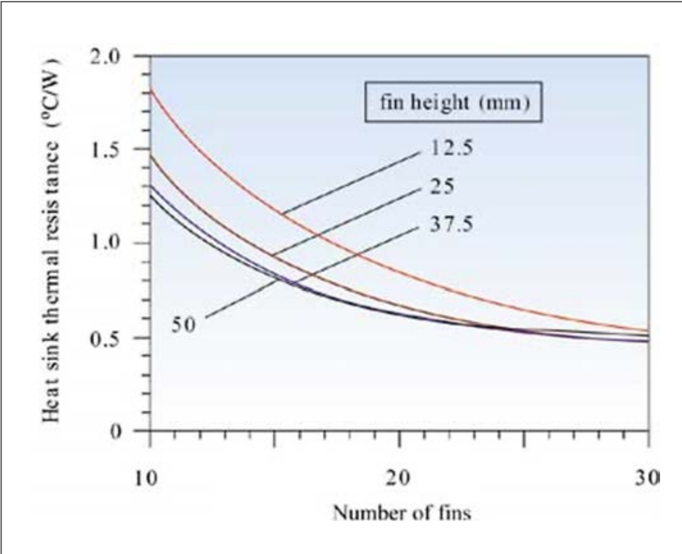

Figure 3: Effect of fin height and number of fins on heat sink thermal resistance at a volumetric air flow rate of 0.0024 m^3/s (5 CFM).

For purposes of illustration these equations were used to estimate heat sink thermal resistance for a 50 x 50 mm aluminum heat sink. The effect of increasing the fin height and the number of fins is shown in *Figure 2* for a constant air velocity, and in *Figure 3* for a constant volumetric flow rate. In both cases it may be seen that there are limits to how much heat sink thermal resistance may be reduced by either increasing fin length or adding more fins. Of course, to determine how a heat sink will actually perform in a specific application it is necessary to determine the air velocity or volumetric flow rate that can be delivered through the heat sink. To do this, it is necessary to estimate the heat sink pressure drop characteristics and match them to the fan or blower to be used. This is a topic for consideration in a future article.

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Calculating Interface Resistance

Reprinted from May 1997

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****Note:** Affiliation as cited in the original article*

EDITOR'S COMMENTS:
My spreadsheet of useful analysis tools also includes multiple contributions from Mike Yovanovich and his co-workers. This article provides a comprehensive overview of what factors lead to interface resistance and a straightforward method for predicting the effects of materials, surface conditions, pressure, the use of a thermal interface material, etc. I have probably used my calculator that implements the analysis, which is shown in this article, hundreds of times during my career. ***(Ross Wilcoxon)***

INTRODUCTION

The exposed surface area of many of today's high-powered electronic packages is no longer sufficient for the removal of the heat generated during normal operation. Heat sinks are a commonly-used, low cost means of increasing the effective surface area for dissipating heat by means of convective air cooling. While the use of a heat sink lowers the fluid-side thermal resistance, it also introduces an interface resistance across the contact formed between itself and the package case. Under some circumstances, this contact resistance can be substantial, impeding heat flow and reducing the overall effectiveness of the heat sink. *Figure 1* depicts an electronic package heat sink assembly which would typically be joined by plastic or metal spring clips around the perimeter of the assembly.

Figure 1: Ceramic Package–Aluminum Heat Sink Assembly

The subject of thermal resistance at interfaces between aluminum heat sinks and ceramic packages has been discussed by Lee [1], de Sorgo [2], Latham [3], and Early et al. [4]. These articles primarily report test results for joint resistance as a function of contact pressure for various interface types. The interfaces examined in these works involve either bare surfaces (air filled) or joints where the interstitial gap is filled with a material layer containing dispersed thermally conductive fillers. Interstitial material layers currently used by the industry, as described by de Sorgo [2], include thermal greases, thermally conductive compounds, elastomers, and adhesive tapes.

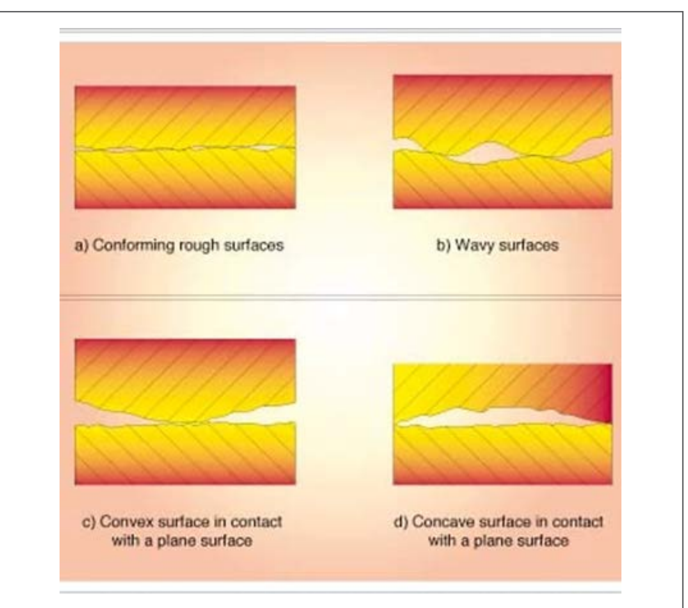

Figure 2: Contact Configurations

The objective of this article is to illustrate how to calculate the ther-

mal joint resistance for the interface formed by two conforming, rough surface shown in *Figure 2a*, as a function of contact pressure for the low pressure range, between 0.035 and 0.35 *MPa* (5 and 50 psi), commonly encountered in microelectronic applications (Latham [3]). Peterson and Fletcher [5] verified by experiments in vacuum that the following models, which were originally developed for metal-to-metal contacts, give very good results when used to predict the contact conductance at interfaces formed by metals (invar, Kovar, and alloy 42) and mold compounds (Polyset 410B and 410C, MG25F-LMP, and MG45F-04) at the interface temperature range: 20°C to 70°C and the interface pressure range: 0.5 to 5.0 *MPa*. This work will focus primarily on bare joints, although an example where the interface material is treated as a liquid, such as in the case of a thermal grease, will also be considered. Interfaces with thermal compounds or elastomeric sheet materials will be shown to be very difficult to model and will be discussed in general terms only. The non-conforming wavy, convex, or concave interfaces depicted in *Figures 2b*, *2c*, and *2d*, respectively, are exceedingly complex to model and therefore will not be considered here. Since radiation heat transfer at most interfaces is negligible or non-existent, it will not be included in this analysis.

CONFORMING ROUGH SURFACE MODEL

The thermal joint conductance, h_j, of the interface formed by two conforming, rough surfaces is given by the following simple model proposed by Yovanovich [6] and further described and used by Antonetti and Yovanovich [7], Yovanovich and Antonetti [8], and Yovanovich [9].

$$h_j = h_c + h_g$$

The contact conductance is given by:

$$h_c = 1.25 k_s \frac{m}{\sigma} \left(\frac{P}{H_c}\right)^{0.95}$$

where k_s is the harmonic mean thermal conductivity of the interface:

$$k_s = \frac{2k_1 k_2}{k_1 + k_2}$$

the effective mean absolute asperity slope of the interface m, as shown in *Figure 3*, is given by:

$$m = \sqrt{m_1^2 + m_2^2}$$

and where σ, also shown in *Figure 3*, is the effective RMS surface roughness of the contacting asperities:

$$\sigma = \sqrt{\sigma_1^2 + \sigma_2^2}$$

The contact pressure is P and H_c is the surface microhardness of the softer of the two contacting solids. The microhardness is in general complex because it depends on several geometric and physical parameters, such as the Vickers microhardness correlation coefficients.

The surface asperity slope is frequently not given. In this case, the mean absolute asperity slope can be approximated by the correlation equation proposed by Antonetti et al. [10]:

$$m = 0.125(\sigma * 10^6)^{0.402}$$

which was developed for the surface roughness range:

$$0.216\ \mu m \leq \sigma < 9.6\ \mu m$$

The gap conductance, h_g, is given by the approximation of Yovanovich [6]:

$$h_g = \frac{k_g}{Y + M}$$

where k_g is the thermal conductivity of the gap substance. The effective gap thickness Y, shown in *Figure 3*, can be calculated accurately by means of the simple power-law correlation equation proposed by Antonetti and Yovanovich [7]:

$$Y = 1.53\sigma \left(\frac{P}{H_c}\right)^{-0.097}$$

for the relative contact pressure range:

$$10^{-5} < P/H_c < 2 \times 10^{-2}$$

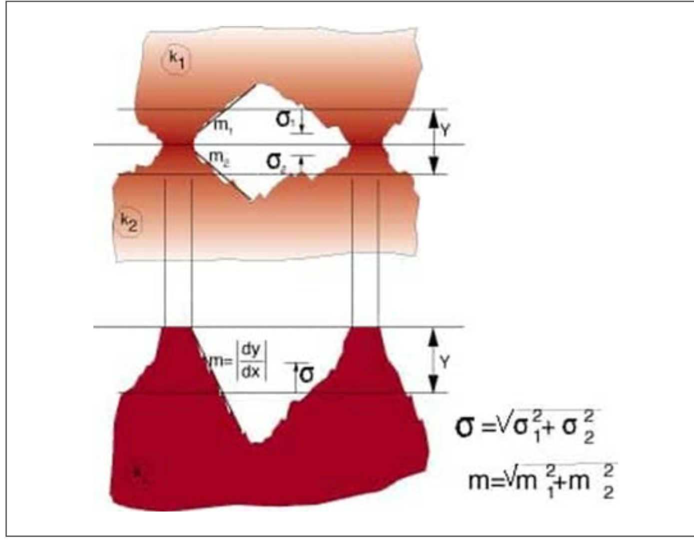

Figure 3: Conforming rough surfaces

The gas parameter M accounts for rarefaction effects at high temperatures and low gas pressures. This gas-surface parameter depends on the thermal accommodation coefficients, the ratio of specific heats, the Prandtl number, and the molecular mean free-path of the gas. Song and Yovanovich [11] present correlation

equations for the calculation of the accommodation coefficients for several gases as a function of the gas temperature. This complex gas-surface parameter depends on gas pressure and temperature according to the relationship:

$$M = M_0 \frac{T}{T_0} \frac{P_{g,0}}{P_g}$$

where M_0 denotes the gas parameter value at the reference values of gas temperature and pressure, T_0 and $P_{g,0}$, respectively. Reference values of the gas parameter for air and helium are presented in *Table 1*.

Table 1		
Gap Substance	Thermal Conductivity (W/mK)	Gas Parameter $M_0 \times 10^6$, m
air	0.026	0.373**
helium	0.150	2.05**
thermal grease	0.20 – 0.70*	0.0
doped thermal grease	1.68 – 2.58*	0.0
** – T0 – 50°C, Pg,0 = 1 atm * – AOS Technical Data Sheets, 1995		

INTERSTITIAL MATERIAL LAYERS

Although the conforming rough surface model presented in the previous section was developed for bare surfaces, it can also be applied to interfaces with thermal grease. By assuming that the grease behaves as a liquid and fills all gaps between the contacting asperities, the existing model can be used by substituting M = 0 and the thermal conductivity of the grease into the gap conductance relationship. However, when solid interstitial materials are used, such as thermal compounds, elastomers or adhesive tapes, the joint conductance problem becomes much more complicated. As shown in *Figure 4*, the use of a solid interstitial material introduces an additional interface to the problem.

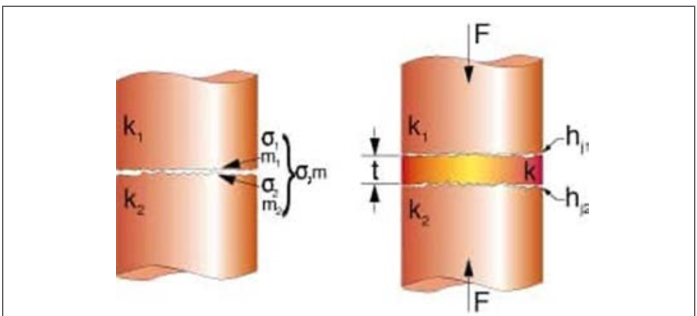

Figure 4: Thermal conductance across an interface with and without an interstitial material

Using thermal resistance concepts, the overall joint conductance for this problem is determined by the series combination:

$$\frac{1}{h_j} = \frac{1}{h_{j,1}} + \frac{t}{k} + \frac{1}{h_{j,2}}$$

where $h_{j,1}$ and $h_{j,2}$ refer to the joint conductance between each of the contacting surfaces and the interfacial material and t and k are the average thickness and thermal conductivity of the layer. Completing this analysis requires characterization of the relevant surface parameters, such as the slope, roughness and microhardness, for the various interstitial materials. In addition, for elastomeric materials the layer thickness t is not constant but instead depends on the contact pressure. Additional research needs to be done before a model can be developed to address this complex phenomenon.

APPLICATION TO ALUMINUM HEAT SINK-CERAMIC PACKAGE INTERFACE

Table 2: Thermal and Surface Properties for Aluminum-Alumina Conforming Rough Surfaces			
Material	Thermal Conductivity (W/mK)	Microhardness MPa	Surface Roughness µm
A1 5052 [14]	140	745	6.9
A1 6061 [14]	180	705	0.7
A1 6063-T5	201	1094	0.4 (flycut)
Aluminum Nitride [13]	160	10044	0.45
Alumina (96% A1203)	20.9	3100	1.3 (ground)
Copper [13]	397	924.1	0.45 (milled)

The aforementioned models will be used to calculate the joint resistances for the interface formed by an aluminum 6063-T5 aluminum heat sink and Al_2O_3alumina package. The thermal conductivities of the heat sink and ceramic package are k_1= 201 W/mK and k_2 = 20.9 W/mK respectively. The harmonic mean thermal conductivity of the interface is k_s = 37.85 W/mK. Since the microhardness of the aluminum alloy is 1094 MPa, which is much less than that of the alumina, it will be used to compute the contact parameters. Based on a surface roughness for flycut aluminum of σ_1= 0.4 µm and a surface roughness for ground alumina of σ_2 = 1.3 µm, the effective surface roughness of the interface is calculated as σ = 1.36 µm. Since the surface slopes are not given, Eq. (6) will be used to calculate the following values: m_1 = 0.139, m_2 = 0.0865, respectively. The effective surface slope of the interface is therefore m = 0.164. The thermal and physical properties of air, helium and grease presented in *Table 1* will be used in the gap conductance model.

In *Figure 5* the joint thermal resistances, whose units are cm^2 °C/W are plotted against the nominal contact pressure over the pressure range: $0.007 \leq P$ (MPa) ≤ 0.35 for several cases. The bare joint resistances with air or helium present in the gap are shown. The effect of a thermal grease of thermal conductivity k_g = 0.20 W/mK is also shown in *Figure 5*.

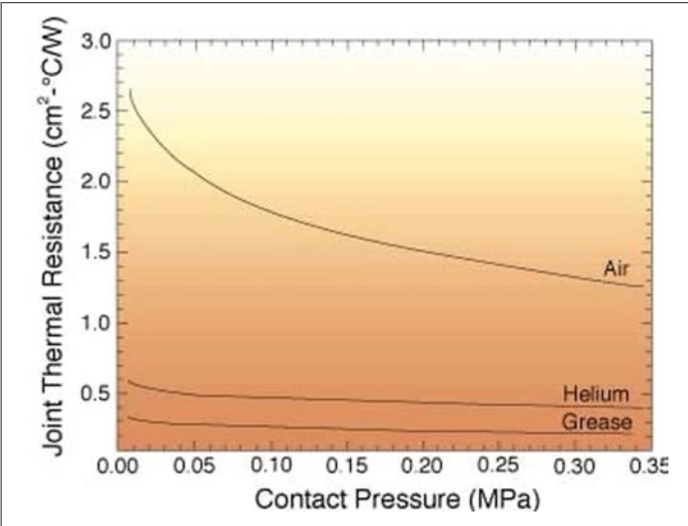

Figure 5: Joint thermal resistance of an aluminum heat sink-ceramic package assembly for various contact pressures.

SUMMARY AND DISCUSSION

Simple correlation equations are presented and used to calculate thermal joint resistances for a typical aluminum-ceramic interface found in microelectronics applications. Flycut and ground surfaces are considered. Joint resistances are calculated for contact pressures between 0.007 and 0.35 MPa, which includes the practical microelectronic pressure range of 0.07 and 0.17 MPa (see Latham [3]). The greatest joint resistances are found when air is present in the interstitial gap. In the contact pressure range of 0.007 to 0.35 MPa, the air joint resistance goes from 2.665 to 1.903 cm^2°C/W.

When silicon grease is placed in the gap, the joint resistance is much smaller than the bare interface. The calculated values of the joint resistance lie in the range 0.335 to 0.213 cm^2°C/W which are an order of magnitude smaller than the joint resistances of a bare joint. If greases with thermally conductive ceramics are used, the joint resistance can be reduced to values below 0.065 cm^2°C/W.

The correlation equations which have been used are based on conforming rough surfaces with interstitial substances which perfectly wet all portions of the surfaces which form the gap. Any non-flatness will result in interfaces with larger gaps which will have larger joint resistances. If the interstitial substance does not perfectly wet the contacting surfaces, this will also produce a more thermally resistive interface. The proposed models and correlation equations therefore correspond to the best thermal joints which have the smallest joint resistances.

The use of other interstitial materials, such as thermal compounds, elastomers or adhesive tapes, has been shown to increase the complexity of the joint conductance problem significantly. In order to successfully model this problem, extensive research into characterizing the surface properties and layer thicknesses for the various interfacial materials is required.

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Calculating Spreading Resistance in Heat Sinks

Reprinted from January 1998

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***Note:** Affiliation as cited in the original article*

EDITOR'S COMMENTS:
A classical study calculating the thermal spreading resistance, used over the years by many practicing engineers to solve the chip to substrate to PCB heat spreading challenges. This valuable material transcends time and is an excellent reference for our thermal community. ***(Victor Chiriac)***

Accident? Consider the scenario where a designer wishes to incorporate a newly developed device into a system and soon learns that a heat sink is needed to cool the device. The designer finds a rather large heat sink in a catalog which marginally satisfies the required thermal criteria. Due to other considerations, such as fan noise and cost constraints, an attempt to use a smaller heat sink proved futile, and so the larger heat sink was accepted into the design. A prototype was made which, unfortunately, burned-out during the initial validation test, the product missed the narrow introduction time, and the project was canceled. What went wrong?

The reasons could have been multi-fold. But, under this scenario, the main culprit could have been the spreading resistance that was overlooked during the design process. It is very important for heat sink users to realize that, unless the heat sink is custom developed for a specific application, thermal performance values provided in vendor's catalogs rarely account for the additional resistances coming from the size and location considerations of a heat source. It is understandable that the vendors themselves could not possibly know what kind of devices the users will be cooling with their products.

INTRODUCTION

Spreading or constriction resistances exist whenever heat flows from one region to another in different cross sectional area. In the case of heat sink applications, the spreading resistance occurs in the base-plate when a heat source of a smaller footprint area is mounted on a heat sink with a larger base-plate area. This results in a higher local temperature at the location where the heat source is placed. *Figure 1* illustrates how the surface temperature of a heat sink base-plate would respond as the size of the heat source is progressively reduced from left to right with all other conditions unchanged: the smaller the heat source, the more spreading has to take place, resulting in a greater temperature rise at the center. In this example, the effect of the edge surfaces of the heat sink is ignored and the heat source is assumed to be generating uniform heat flux.

In cases where the footprint of a heat sink need not be much larger than the size of the heat source, the contribution of the spreading resistance to the overall device temperature rise may be insignificant and usually falls within the design margin. However, in an attempt to remove more heat from today's high performance devices, a larger heat sink is often used and, consequently, the impact of spreading resistance on the performance of a heat sink is becoming an important factor that must not be ignored in the design process. It is not uncommon to find in many high performance, high power applications that more than half the total temperature rise of a heat sink is attributed to the spreading resistance in the base-plate.

The objectives of this article are:

1. To understand the physics and parameters associated with spreading resistance
2. To provide a simple design correlation for accurate prediction of the resistance
3. To discuss and clarify the concept of spreading resistance with an emphasis on the practical use of the correlation in heat sink applications

The correlation provided herein was originally developed in *References [1]* and *[2]*. This article is an extension of the earlier presentation.

SPREADING RESISTANCE

Before we proceed with the analysis, let us attend to what the temperature distributions shown in *Figure 1* are telling us. The first obvious one, as noted earlier, is that the maximum temperature at the center increases as the heat source becomes smaller. Another important observation is that, as the temperature rises in the center, the temperatures along the edges of the heat sink decrease simultaneously. It can be shown that this happens in such a way that the area-averaged surface temperature of the heat sink base-plate has remained the same. In other words, the average heat sink

thermal performance is independent of the size of a heat source. In fact, as will be seen later, it is also independent of the location of the heat source.

The spreading resistance can be determined from the following set of parameters:

- Footprint or contact area of the heat source, A_s
- Footprint area of the heat sink base-plate, A_p
- Thickness of the heat sink base-plate, t
- Thermal conductivity of the heat sink base-plate, k
- Average heat sink thermal resistance, R_0

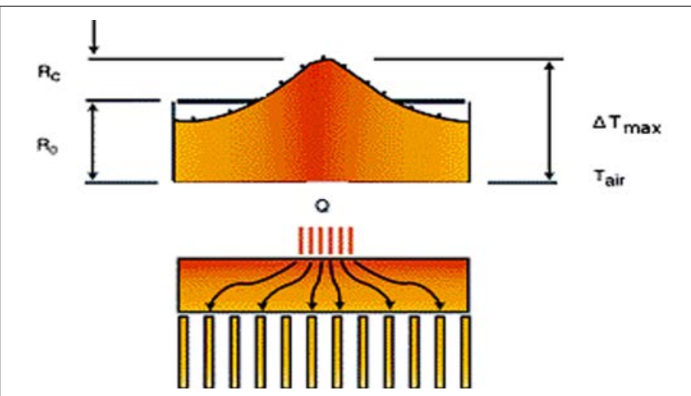
We will assume, for the time being, that the heat source is centrally mounted on the base-plate, and the heat sink is cooled uniformly over the exposed finned surface. These two assumptions will be examined in further detail. *Figure 2* shows a two-dimensional side view of the heat sink with heat-flow lines schematically drawn in the base-plate whose thickness is greatly exaggerated. At the top, the corresponding surface temperature variation across the center line of the base-plate is shown by the solid line. The dotted line represents the average temperature of the surface which is, again, independent of the heat source size and can be easily determined by multiplying R_0 with the total amount of heat dissipation, denoted as Q.

As indicated in *Figure 2*, the maximum constriction resistance R_c, which accounts for the local temperature rise over the average surface temperature, is the only additional quantity that is needed for determining the maximum heat sink temperature. It can be accurately determined from the following correlation.

$$R_c = \frac{\sqrt{A_p}-\sqrt{A_s}}{k\sqrt{\pi A_p A_s}} x \frac{\lambda k A_p R_0 + \tanh(\lambda t)}{1+\lambda k A_p R_0 \tanh(\lambda t)} \quad (1)$$

where

$$\lambda = \pi^{3/2}/\sqrt{A_p} + 1/\sqrt{A_s} \quad (2)$$



Figures 1, 2: Two dimensional schematic view of local resistance or temperature variation of a heat sink shown with heat flow Lines

EXAMPLE PROBLEM

Consider an aluminum heat sink (k = 200 W/mK) with base-plate dimensions of 100 x 100 x 1.3 mm thick. According to the catalog, the thermal resistance of this heat sink under a given set of conditions is 1.0°C/W. Find the maximum resistance of the heat sink if used to cool a 25 x 25 mm device.

SOLUTIONS

With no other specific descriptions, it is assumed that the heat source is centrally mounted, and the given thermal resistance of 1.0°C/W represents the average heat sink performance. From the problem statement, we summarize:

- $A_s = 0.025 \times 0.025 = 0.000625\ m^2$
- $A_p = 0.1 \times 0.1 = 0.01\ m^2$
- t = 0.0013 m
- k = 200 W/mK
- $R_0 = 1.0°C/W$

Therefore,

$$\lambda = \pi^{3/2}/\sqrt{0.01} + 1/\sqrt{0.000625} = 95.68\ m^{-1}$$

$$\tanh(\lambda t) = \tanh(95.68 \times 0.0013) = 0.124$$

$$\lambda k A_p R_0 = 95.68 \times 200 \times 0.01 \times 1.0 = 191.4$$

$$R_c = \frac{\sqrt{0.01}-\sqrt{0.000625}}{200\sqrt{\pi x 0.01 x 0.000625}} x \frac{191.4+0.124}{1+191.4 \times 0.124} = 0.0846 \times 7.743 = 0.66°C/W$$

Hence, the maximum resistance, R_{total}, is:

$$R_{total} = R_o + R_c = 1.0 + 0.66 = 1.66°C/W$$

EFFECT OF SOURCE LOCATION

In the following two sections, we will limit our examination to the current example problem. As we shall see, the result of this limited case study will allow us to draw some general yet useful conclusions. Suppose the same heat source in the above example was not centrally located, but mounted a distance away from the center. Obviously, the maximum temperature would further rise as compared to that found in the above example. *Figure 3* shows the local resistances corresponding to two such cases:

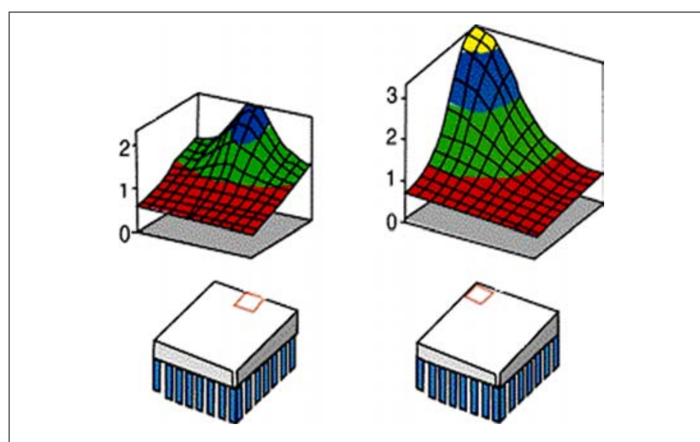


Figure 3: Heat sink local resistance showing the effect of source location

From L to R, heat source at (37.5,0) and (37.5,37.5) the first one is for the case where the heat source is mounted midway along the edge, and the other, where it is mounted on one corner of the heat sink.

For these two special cases, the maximum spreading resistance can be calculated by using *Equation (1)* for R_c with input parameters t and R_0 modified as shown below:

$$R_c = C \times R_c\,(A_p, A_s, k, t/C, R_o/C) \qquad (3)$$

with C = 2^(1/2) for the first case, and C = 2 for the second case. It is to be noted that this expression is independent of the source size. Numerically, for the current problem with a 25 x 25 mm heat source, it results in the maximum spreading resistances of 1.29 and 2.38°C/W, or the total resistances of 2.29 and 3.38°C/W for the first and second cases, respectively. For both cases, it can be shown that the average surface resistance has not changed from unity.

For other intermediate source locations, numerical simulations were carried out and a plot is provided in *Figure 4* for the correction factor C_f, which can be used to compute the total resistance as

$$R_{total} = R_o + C_f R_c \qquad (4)$$

Where: R_c is determined from *Equation (1)*, given for the case with the heat source placed at the center.

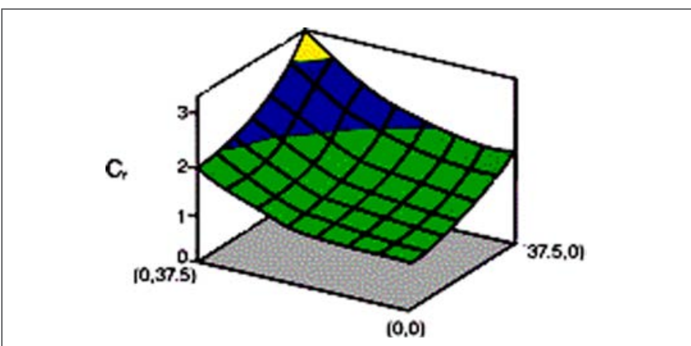


Figure 4: Correction factor as a function of source location

The coordinates in *Figure 4* indicate the location of the center of the heat source measured from the center of the base-plate in mm: the case with a centrally located heat source corresponds to (0,0), and the cases shown in *Figure 3* correspond to (37.5,0) and (37.5,37.5) for the first and second cases, respectively. Only one quadrant is shown in *Figure 4* as they would be, owing to the assumption of uniform cooling, symmetrical about (0,0). As can be seen from the figure, the correction factor increases from 1 as the heat source is placed away from the center. It is worthwhile noting that the increase is, however, very minimal over a wide region near the center, and most increases occur closer to the edges.

Unlike C in the earlier expression, C_f is case dependent (i.e. it depends on the heat-source size). However, it was found that the plots of C_f obtained for many other cases exhibit essentially the same profile as that shown in *Figure 4*, with magnitudes at the corners determined from *Equation (3)*, and the domain of the plot defined by the maximum displacement of the heat source. Based on this observation, a general conclusion can be made: for all practical purposes, as long as the heat source is placed closer to the center than to the edges of the heat sink, the correctional increase in the spreading resistance may be ignored, and C_f =1 may be used. As noted above, this would introduce a small error of no greater than 5-10% in the spreading resistance which, in turn, is a fraction of the total resistance.

So far, we have assumed a uniform cooling over the entire finned-surface area of the base-plate. Although this is a useful assumption, it is seldom realized in actual situations. It is well known that, due to the thinner boundary layer and the less downstream heating effect, a device would be cooled more effectively if it is mounted toward the air inlet side. Again, a numerical simulation is carried out using our example problem with the boundary layer effect included.

Figure 5 shows the resulting modified correction factor as a function of the distance from the center of the heat sink to the heat source placed along the center line at y = 0: x = -37.5 mm corresponds to the front most leading edge location of the heat source and x = 37.5 mm the rear most trailing edge placement.

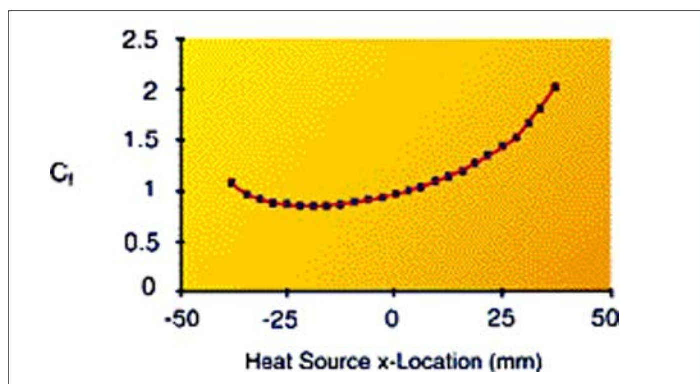


Figure 5: Correction factor modified for boundary layer effect at y=0

As can be seen from the figure, it is possible to realize a small improvement by placing the heat source forward of the center location where $C_f < 1$. However, it was experienced in practice that accommodating a heat source away from the center and ensuring its mounting orientation often cause additional problems during manufacturing and assembly processes.

SUMMARY AND DISCUSSION

A simple correlation equation is presented for determining spreading resistances in heat sink applications. A sample calculation is carried out for a case with a heat source placed at the center of the heat sink base-plate and a means to estimate the correction factor to account for the effect of changing the heat source location is provided. It is to be noted that the correlation provid-

ed herein is a general solution which reduces to the well-known Kennedy's solution [3] when R_0 approaches 0: the mathematical equivalent of isothermal boundary condition. Kennedy's solution is valid only when R_0 is sufficiently small such that the fin-side of the heat sink base-plate is close to isothermal. Otherwise, Kennedy's solution, representing the lower boundary of the spreading resistance, may result in gross underestimation of the resistance.

The earlier study revealed that, depending on the relative magnitude of the average heat sink resistance, the spreading resistance may either increase or decrease with the base-plate thickness. If the heat sink resistance is sufficiently small, as in liquid cooled heat sink applications, the spreading resistance always increases with the thickness, and an optimum thickness does not exist. On the other hand, if the heat sink resistance is large, as experienced in most air-cooled applications, the spreading resistance decreases with the thickness and a finite optimum thickness exists.

It is to be noted that the present correlation calculates the spreading resistance only in the base-plate and does not account for the effect of additional spreading that may exist in other places, such as the fins in a planar heat sink. This additional spreading in the fins usually affects the spreading resistance in a similar way to a thicker base-plate. The current author found that an increase of 20% in the base-plate thickness during the calculation roughly accounts for the effect of this additional spreading in the fins of the same material for most planar heat sinks under air cooling. No modification is required for pin-fin heat sinks.

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Thermal Interface Materials: A Brief Review of Design Characteristics and Materials

Reprinted from February 2004

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EDITOR'S COMMENTS:
This article provides a review of the thermal interface materials, a topic of significant interest to our thermal community. The study includes a comprehensive study of the types and the main features and characteristics of the thermal interface materials: the thermal conductivity, the bond line thickness (BLT), the bulk and the contact resistances, also various useful correlations and comparisons to empirical models. A valuable study for the practicing thermal engineer. ***(Victor Chiriac)***

INTRODUCTION

In the last few decades, as microprocessors have continued to evolve along Moore's law, providing increased functionality and performance, there has been an associated increase in cooling demand driven both by the increase in raw power and in local power densities on the die, commonly referred to as "hot spots"[1, 2]. Considerable attention has therefore been given to the development of thermal solutions and to the characterization of all aspects of the thermal solution. The scope of the development has been constrained significantly by cost pressures and the desire to develop solutions that meet stringent form factor constraints as the marketplace has seen an evolution of form factors increasing the form, fit, and function demands on the thermal solutions. In parallel, the relentless pace of microprocessor evolution implies that solution developers must also deal with limited time schedules.

In light of the environmental pressures described above, industrial and academic researchers have partnered well to effectively bring thermal solutions to the marketplace, while, in parallel, systematically investigating the fundamental aspects of thermal solution development. As discussed in [2, 3], a simple classification of thermal solutions may be made into two architectures, schematically illustrated in *Figure 1*. A key assumption here is that the bulk of the heat transfer occurs through the inactive side of the silicon and that thermal management addresses this primary heat transfer path.

In the first architecture, a heat sink is typically applied directly to the back side of a silicon device through a thermal interface material (TIM). Such architectures are often used in space-constrained applications, such as laptops. The second architecture has a separate heat spreader interfacing directly with the inactive side of the die, and a heat sink interfaces to the heat spreader. In this architecture there are two separate applications of the TIM. It is important to note that in both architectures, the TIM plays a key role in connecting the different aspects of the solution (i.e., the heat source [the die]) with the heat-spreader and/or the heat sink, ensuring efficient transfer of heat. The focus of this article is to discuss the desired characteristics of the TIM, the different classes of TIMs, and their advantages and limitations.

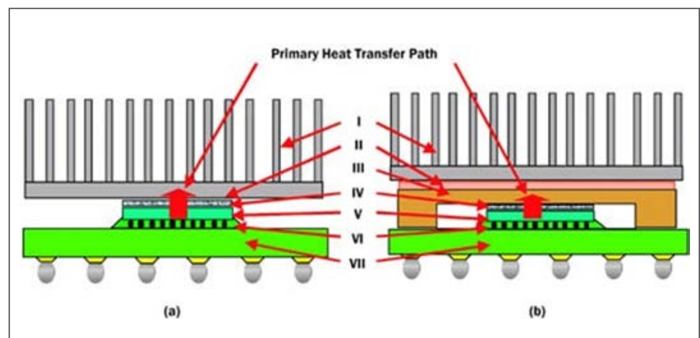

Figure 1: Schematic illustration of the two thermal architectures:
(a) Architecture I is typically used in laptop applications.
(b) Architecture II is typically used in desktop and server applications.
(Legend: I – Heat Sink, II – TIM, III – IHS, IV – TIM, V – Die, VI – Underfill, and VII – Package Substrate)

DESIRED CHARACTERISTICS OF A TIM

As discussed in the introduction, the TIM acts to connect the different parts of a thermal solution. After inserting a TIM between the solid surfaces, the effective thermal resistance, R_{TIM}, at the interface will have two components, i.e., the bulk resistance, R_{bulk}, of the TIM arising from its finite thermal conductivity and the contact resistance, R_c between the TIM and the adjoining solids, as shown in *Figure 2*. R_{TIM} may be expressed as:

$$R_{TIM} = \frac{BLT}{k_{TIM}} + R_{c_1} + R_{c_2} \quad (1)$$

where BLT is the bond-line thickness of the TIM, k_{TIM} is the thermal conductivity of the TIM, and R_{c1} and R_{c2} are the contact resistances of the TIM with the two adjoining surfaces. Note that R_{TIM}, R_{c1}, and R_{c2} are area-normalized thermal resistances (K-m²/W). One of the goals of thermal design is to reduce R_{TIM}. This can be accomplished by reducing the BLT, increasing the thermal conductivity and reducing the contact resistances R_{c1} and R_{c2}. Let us examine in detail how each of these may be accomplished.

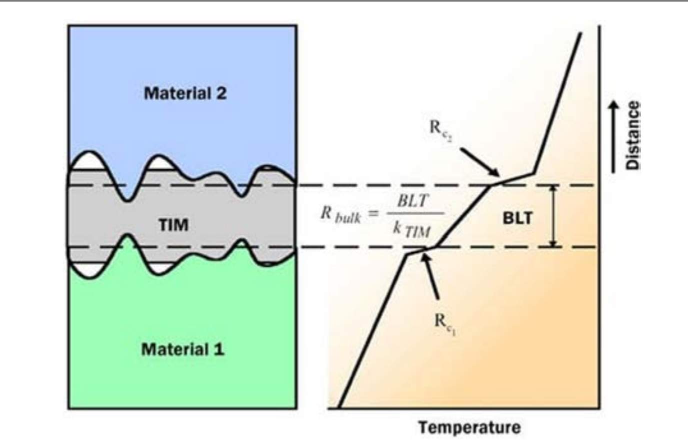

Figure 2: Schematic of various resistance components of R_{TIM}.

THERMAL CONDUCTIVITY

In typical applications [4, 5], the TIM serves to conduct heat through its thickness. The thermal conductivity of a TIM is typically enhanced by loading a soft, sometimes liquid-like polymeric material matrix with conducting solid particles, such as aluminum, alumina, and boron nitride. If the design requirement is that the TIM should be thermally conducting but electrically insulating then the ceramic-based filler particles are more typically chosen. *Figure 3* shows the variation of thermal conductivity of silicone based thermal grease as a function of filler (aluminum) particle volume fraction [6] where km is the thermal conductivity of the silicone oil.

Thermal conductivity of particle laden TIMs can be functionally expressed as:

$$k_{TIM} = f(k_f, k_m, \phi, R_b) \tag{2}$$

Where k_f is the filler thermal conductivity, Φ, filler volume fraction, and R_b is the contact resistance between the fillers and the polymeric matrix. There are various analytical models available in the literature [6]. For spherical particles, one of the most prominent models is the Maxwell model. This model matches the data for spherical particles with Φ up to 30%-35%, after which percolation phenomenon takes over. Maxwell's model cannot be used to predict the thermal conductivity for higher volume fraction due to the assumptions built into it. Prasher et al. in [6] used the modified Bruggeman model due its ability to predict thermal conductivity from low to high volume fraction and also because it includes the effect of the interface resistance between the filler and the matrix on the effective thermal conductivity of composites. The modified Bruggeman model for $K_{TIM}/k_m >>1$ is given by:

$$\frac{k_{TIM}}{k_m} = \frac{1}{(1-\phi)^{3(1-\alpha)/(1+2\alpha)}} \tag{3}$$

where α is the Biot number and is given as:

$$\alpha = \frac{R_b k_m}{d} \tag{4}$$

where R_b is the interface resistance between the filler and the particle and d is the diameter of the particle. Note that R_b is expressed in units of area-normalized thermal resistance (Km²/W). The slid line in *Figure 3* represents the results of the modified Bruggeman model for aluminum filler laden silicon-based grease for different values of α. Thermal conductivity of silicone was assumed to be 0.2 W/mK. *Figure 3* shows that model matches very well with 40% and 50% volume fraction with α = 0.06 and for 60% volume fraction with α = 0.15. This suggests that interface resistance for 60% sample is higher than 40% and 50% samples. R_b could arise because of imperfect mixing of the particle with the polymer matrix, or due to phonon acoustic mismatch, or due to a combination of the two phenomenon.

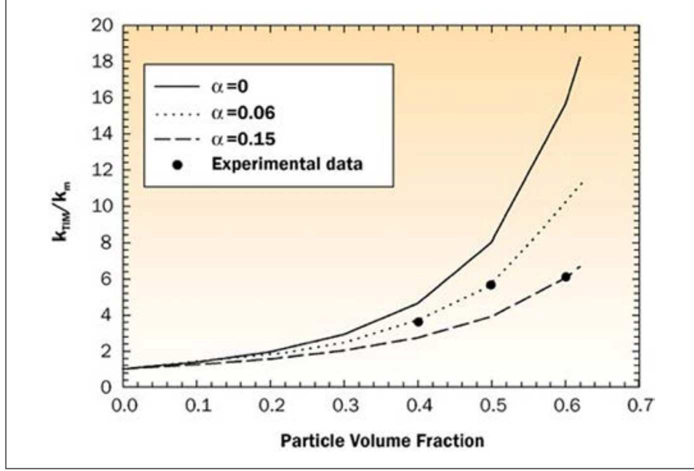

Figure 3: Thermal conductivity vs. particle volume fraction for a silicone based thermal grease with aluminum fillers.

Prasher et al. [6] have shown that the contribution due to phonon acoustic mismatch at room temperatures or higher is important only when the thermal conductivity of both the materials is very high. In the present case, it is safe to assume the R_b is arising from imperfect wetting or mixing of the filler particles with the silicon oil, as the thermal conductivity of silicon oil is very low. Therefore the phonon acoustic mismatch component is very low at room temperatures. The reason for higher R_b for 60% volume fraction could be due to difficulty in wetting the surface of the particle with the silicone oil as the volume fraction is so high.

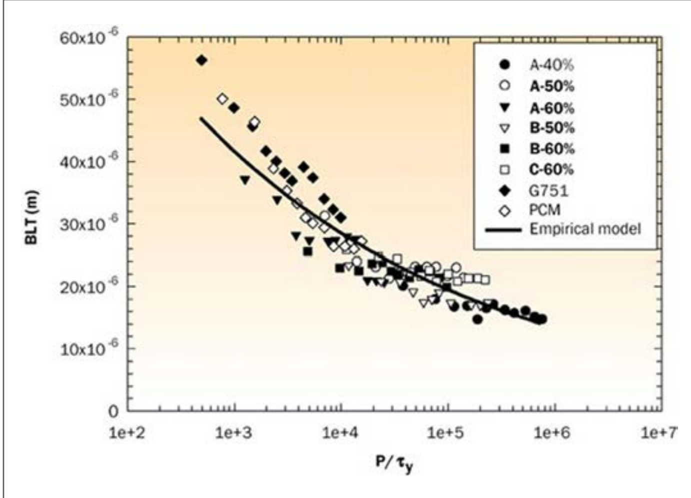

Figure 4: BLT vs. P/τ_y for various TIM materials.

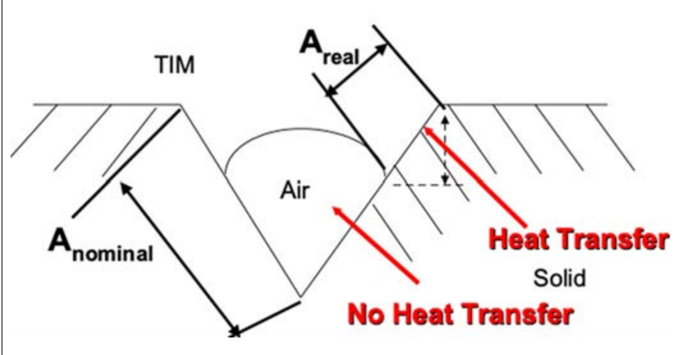

Figure 5: Mechanism of heat transfer near the TIM substrate interface.

BOND-LINE THICKNESS (BLT)

BLT reduction is often another parallel goal of thermal design. BLT is a function of various parameters such as application pressure (i.e., pressure applied in bringing the two contact surfaces together) and particle volume fraction. Prasher et al. [6] have developed an empirical model for the BLT of particle laden polymeric thermal interface materials. They conducted a study on eight different formulations of particle laden polymeric TIMs, which included thermal greases and phase change materials and proposed the following correlation for BLT:

$$BLT = 1.31 \cdot 10^{-4}\left(\frac{\tau_y}{P}\right)^{0.166} \quad (5)$$

where τ_y is the yield stress of the TIM and P is the applied pressure. This correlation was validated in the pressure range of 25-200 psi application pressures. *Figure 4* shows their results. Since the yield stress of the TIM increases with increasing filler loading, BLT is higher for higher volume fraction. Therefore there are two competing effects with regard to filler loading for the thermal resistance of the TIM: k_{TIM} increases and BLT also increases with increasing filler volume fraction at the same pressure, which leads to an optimal filler loading for the minimization of R_{TIM} [6].

CONTACT RESISTANCE

Prasher [8] showed that sum of contact resistance of TIMs with the two adjoining substrates can be written as:

$$R_{c_{1+2}} = \left(\frac{\sigma_1+\sigma_2}{2k_{TIM}}\right)\left(\frac{A_{nominal}}{A_{real}}\right) \quad (6)$$

where σ_1 and σ_2 are the surface roughness of the substrates, $A_{nominal}$ is the nominal area of heat transfer, and A_{real} is the real area of heat transfer. As shown in *Figure 5*, the real area of heat transfer is smaller than the nominal area because of the trapped air in the valleys of a microscopically rough surface of the substrates. In [7] A_{real} was calculated by performing a force balance due to: 1) applied pressure, 2) capillary force due to the surface tension of the TIM, and 3) back pressure due to the trapped air.

Prasher [7] compared the model for phase change and grease type TIMs. *Figure 6* shows a comparison between the model and experimental data on phase change material. Based on this, a few general design guidelines were proposed to minimize contact resistance; i.e., (1) increase pressure, (2) decrease surface roughness, (3) increase thermal conductivity of the TIM, and (4) increase capillary force by changing the surface chemistry.

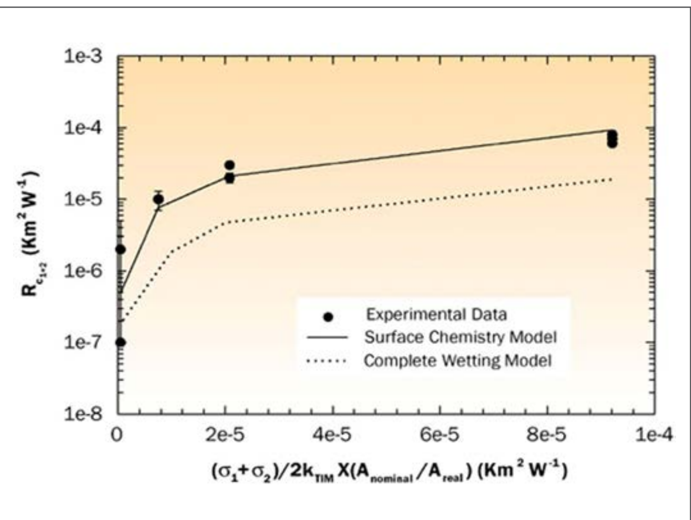

Figure 6: Comparison of the surface chemistry model with experimental results for phase change materials.

RELIABILITY OF TIMS IN TEMPERATURE CYCLING

In packages using thermal grease as the conducting medium between the die and the thermal solution, grease pump-out during operation of the part is a known failure mechanism [8]. Traditionally the power cycle test is a direct method to examine grease reliability. However, it is a time consuming process due to its long heating and cooling times. In order to screen numerous thermal grease materials during the initial design phase of a microprocessor package, it is advantageous to utilize a quick turn test.

As described in [8], an accelerated mechanical test was developed to evaluate interface degradation, due to pump-out. An MTS universal testing machine was used to simulate the squeezing action on the grease, caused by die warpage change.

Figures 7a and *7b* show the typical grease pump-out pattern and temperature trend observed in the quick test, respectively. By using this accelerated testing method, the grease pump-out phenomena can be predicted so that product design time can be significantly reduced.

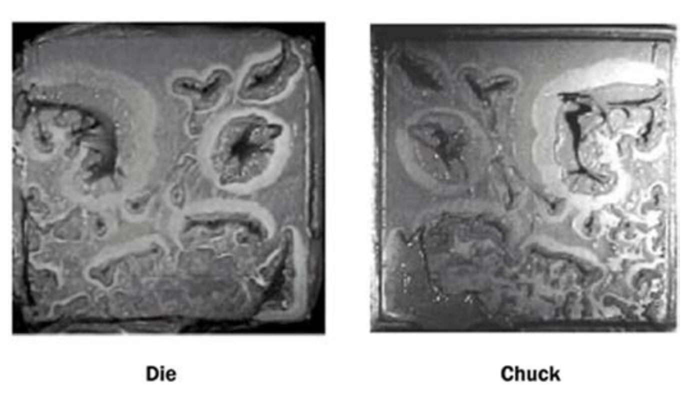

Figure 7a: Post-test thermal grease pattern for the case with grease pump-out.

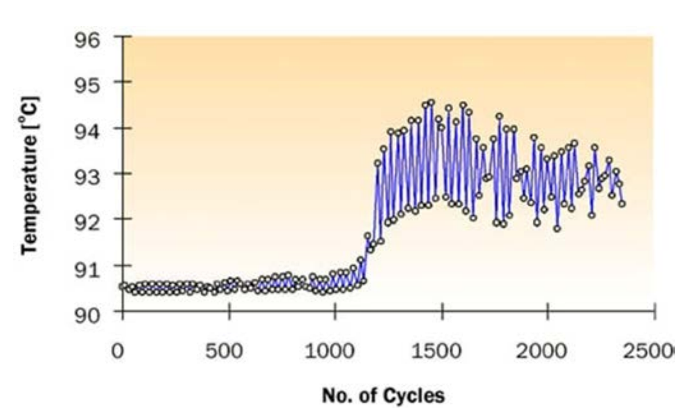

Figure 7b: Temperature trend for a case with grease pump-out.

To solve the pump-out problem gel TIMs were recently developed. Gels are also thermal greases but they cure due crosslinking of the polymer at high temperatures. Recently Prasher and Matayabas [9] proposed the following design rules for the formulation of TIMs to avoid pump out problems and also to have a low thermal resistance:

(a) Minimization of G, where G is given by

$$G = \sqrt{G'^2 + G''^2} \tag{7}$$

where G' and G" are the storage and loss modulus of the polymeric TIM, and

(b) Keep the ratio of G' and G" greater than or equal to 1. *Figure 8* shows the results on the degradation of thermal resistance per cycle from Prasher and Matayabas [9] on eight different samples of thermal greases.

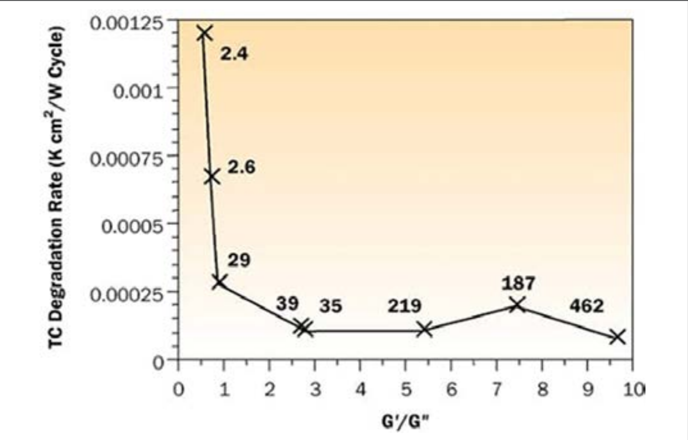

Figure 8: Effect of G'/G" on the degradation rate, as measured by thermal performance of gel TIMs subjected to temperature cycling.

The labels present the value of G for each sample. The data labels in *Figure 8* are the values of G for different samples. So far the discussion has focused on the desired design characteristics of the TIM. These need to be characterized experimentally. Over the last few years, a number of techniques have been elucidated and the researcher today can rely on the hot and cold plate steady-state method [10], a thermal test vehicle in steady-state test [11], a transient test [12], and sandwich-type samples by the laser flash method [13].

RE-WORKABILITY

Another typical requirement of TIMs is that of re-workability. Since in a number of applications, the heat sink is attached to the device by the OEM, re-workability is a requirement to avoid yield loss due to heat sink attach. Re-workability implies that the heat sink should be easily removed and that the TIM should be easily cleanable so that the heat sink may be reattached if needed. This requirement has led to certain classes of materials seeing continued popularity; e.g., filled greases, filled phase change materials, and certain gels.

CLASSES OF TIM MATERIALS

Table 1 shows the characteristics of some of the typical TIM materials and their advantages and disadvantages.

SUMMARY

In this paper, a general overview of the issues that are important in TIM design and the desired characteristics of TIMs have been presented. It was demonstrated that important attributes of a TIM are its thermal conductivity, and its interactions with the mating surfaces; i.e., the resultant BLT and wetting characteristics. Issues, such as reworkability and reliability, also play important roles in the final selection of the TIM.

Table 1: Summary of Characteristics of Some Typical TIM Materials			
TIM Type	General Characteristics	Advantages	Disadvantages
Greases	Typically silicone based matrix loaded with particles (typically AlN or ZnO) to enhance thermal conductivity	• High bulk thermal conductivity • Thin BLT with minimal attach pressure • Low Viscosity enables matrix material to easily fill surface crevices • TIM curing not required • TIM delamination is not a concern	• Susceptible to grease pump-out and phase separation • Considered messy in a manufacturing environment due to a tendency to migrate
Phase Change Materials	Polyolefin, epoxy, low molecular weight polyesters, acrylics typically with BN or Al_2O_3 filters	• Higher viscosity leads to increased stability and hence less susceptible to pumpout • Application and handling is easier compared to greases • No cure required • Dealamination is not a concern	• Lower thermal conductivity than greases • Surface resistance can be greater than greases. Can be reduced by thermal pre-treatment • Requires attach pressure to increase thermal effectiveness
Gels	Al, Al_2O_3, Ag particles in silicone, olefin matrices that require curing	• Conforms to surface irregularity before cure • No pump out of migration concerns	• Cure process needed • Lower thermal conductivity than grease • Lower adhesion than adhesives; delamination can be a concern
Adhesives	Typically Ag particles in a cured epoxy matrix	• Conform to surface irregularity before cure • No pump out • No migration	• Cure process needed • Delamination post reliability testing is a concern • Since cured epoxies have high post cure modules, CTE mismatch induced stress is a concern

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FEATURED

Heat Pipes for Electronics Cooling Applications

Reprinted from September 1996

Scott D. Garner
*Thermacore Inc.
***Note:** *Affiliation as cited in the original article*

EDITOR'S COMMENTS:
This article is one of the most read articles by our community readers. It features a comprehensive overview of the operation, the design, the limits, and the performance of heat pipes. This passive technology is the most efficient means of heat transportation at relatively low cost. Over the last decades, heat pipe designs have evolved to be thinner (flat heat pipes) and to provide heat spreading capabilities (vapor chamber). They are widely used in applications ranging from consumer to IGBT. ***(Genevieve Martin)***

INTRODUCTION

All electronic components, from microprocessors to high end power converters, generate heat and rejection of this heat is necessary for their optimum and reliable operation. As electronic design allows higher throughput in smaller packages, dissipating the heat load becomes a critical design factor. Many of today's electronic devices require cooling beyond the capability of standard metallic heat sinks. The heat pipe is meeting this need and is rapidly becoming a mainstream thermal management tool.

Heat pipes have been commercially available since the mid 1960's. Only in the past few years, however, has the electronics industry embraced heat pipes as reliable, cost-effective solutions for high end cooling applications. The purpose of this article is to explain basic heat pipe operation, review key heat pipe design issues, and to discuss current heat pipe electronic cooling applications.

HEAT PIPE OPERATION

A heat pipe is essentially a passive heat transfer device with an extremely high effective thermal conductivity. The two-phase heat transfer mechanism results in heat transfer capabilities from one hundred to several thousand times that of an equivalent piece of copper.

As shown in *Figure 1*, the heat pipe in its simplest configuration is a closed, evacuated cylindrical vessel with the internal walls lined with a capillary structure or wick that is saturated with a working fluid. Since the heat pipe is evacuated and then charged with the working fluid prior to being sealed, the internal pressure is set by the vapor pressure of the fluid.

As heat is input at the evaporator, fluid is vaporized, creating a pressure gradient in the pipe. This pressure gradient forces the vapor to flow along the pipe to a cooler section where it condenses giving up its latent heat of vaporization. The working fluid is then returned to the evaporator by the capillary forces developed in the wick structure.

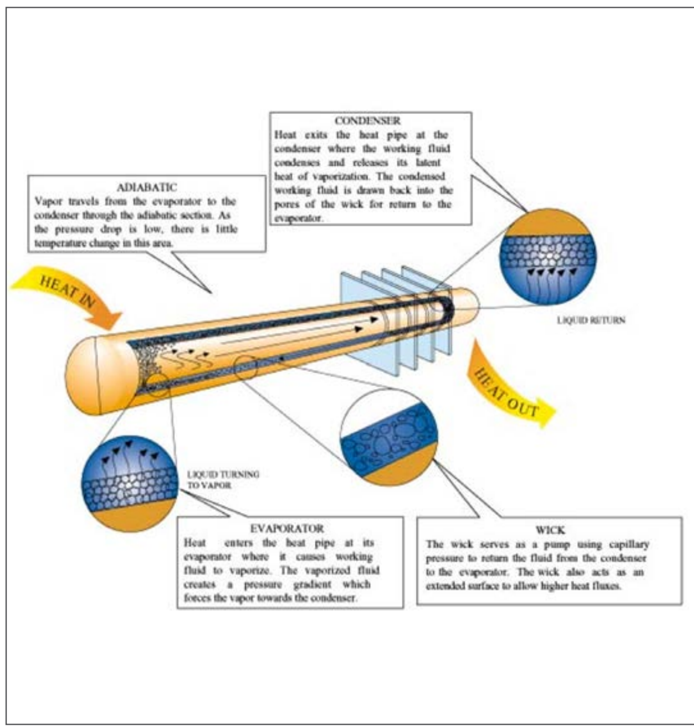

Figure 1: Heat pipe operation

Heat pipes can be designed to operate over a very broad range of temperatures from cryogenic (< - 243°C) applications utilizing titanium alloy/nitrogen heat pipes, to high temperature applications (>2,000°C) using tungsten/silver heat pipes. In electronic cooling applications where it is desirable to maintain junction temperatures below 125-150°C, copper/water heat pipes are typically used. Copper/methanol heat pipes are used if the application requires heat pipe operation below 0°C.

HEAT PIPE DESIGN

There are many factors to consider when designing a heat pipe: compatibility of materials, operating temperature range, diameter, power limitations, thermal resistances, and operating orientation. However, the design issues are reduced to two major considerations by limiting the selection to copper/water heat pipes for cooling electronics. These considerations are the amount of power the heat pipe is capable of carrying and its effective thermal resistance. These two major heat pipe design criteria are discussed below.

LIMITS TO HEAT TRANSPORT

Heat Transport Limit	Description	Cause	Potential Solution
Viscous	Viscous forces prevent vapor flow in the heat pipe	Heat pipe operating below recommended operating temperature	Increase heat pipe operating temperature or find alternativeworking fluid
Sonic	Vapor flow reaches sonic velocity when exiting heat pipeevaporator resulting in a constant heat pipe transport power and largetemperature gradients	Power/temperature combination, too much power at lowoperating temperature	This is typically only a problem at start-up. The heat pipewill carry a set power and the large ^T will self correct as the heat pipe warmsup
Entrainment/Flooding	High velocity vapor flow prevents condensate from returningto evaporator	Heat pipe operating above designed power input or at too lowan operating temperature	Increase vapor space diameter or operating temperature
Capillary	Sum of gravitational, liquid and vapor flow pressure dropsexceed the capillary pumping head of the heat pipe wick structure	Heat pipe input power exceeds the design heat transportcapacity of the heat pipe	Modify heat pipe wick structure design or reduce power input
Boiling	Film boiling in heat pipe evaporator typically initiates at5-10 W/cm² for screen wicks and 20-30 W/cm² for powdermetal wicks	High radial heat flux causes film boiling resulting in heatpipe dryout and large thermal resistances	Use a wick with a higher heat flux capacity or spread outthe heat load

Table 1: Heat pipe heat transport limitations

The most important heat pipe design consideration is the amount of power the heat pipe is capable of transferring. Heat pipes can be designed to carry a few watts or several kilowatts, depending on the application. Heat pipes can transfer much higher powers for a given temperature gradient than even the best metallic conductors. If driven beyond its capacity, however, the effective thermal conductivity of the heat pipe will be significantly reduced. Therefore, it is important to assure that the heat pipe is designed to safely transport the required heat load.

The maximum heat transport capability of the heat pipe is governed by several limiting factors which must be addressed when designing a heat pipe. There are five primary heat pipe heat transport limitations. These heat transport limits, which are a function of the heat pipe operating temperature, include: viscous, sonic, capillary pumping, entrainment or flooding, and boiling. *Figures 2* and *3* show graphs of the axial heat transport limits as a function of operating temperature for typical powder metal and screen wicked heat pipes. Each heat transport limitation is summarized in *Table 1*.

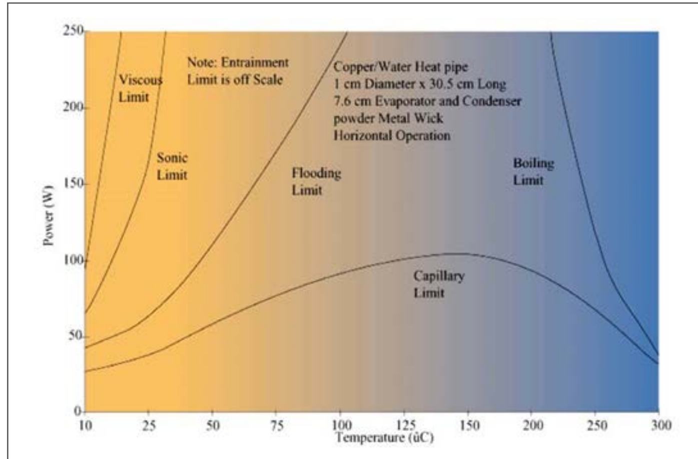

Figure 2: Predicted heat pipe limitations

As shown in *Figures 2* and *3*, the capillary limit is usually the limiting factor in a heat pipe design.

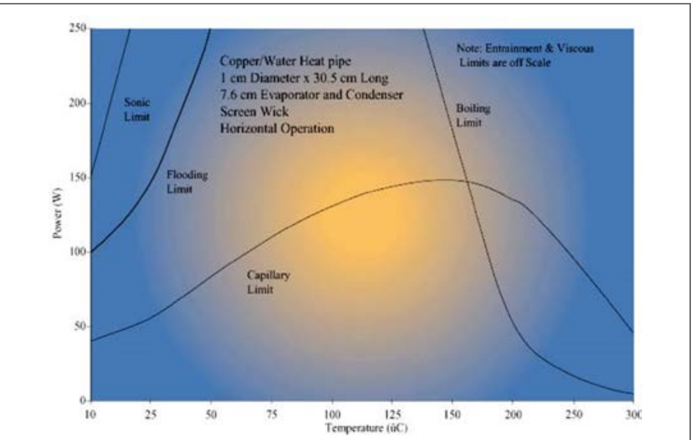

Figure 3: Predicted heat pipe limits

The capillary limit is set by the pumping capacity of the wick structure. As shown in *Figure 4*, the capillary limit is a strong function of the operating orientation and the type of wick structure.

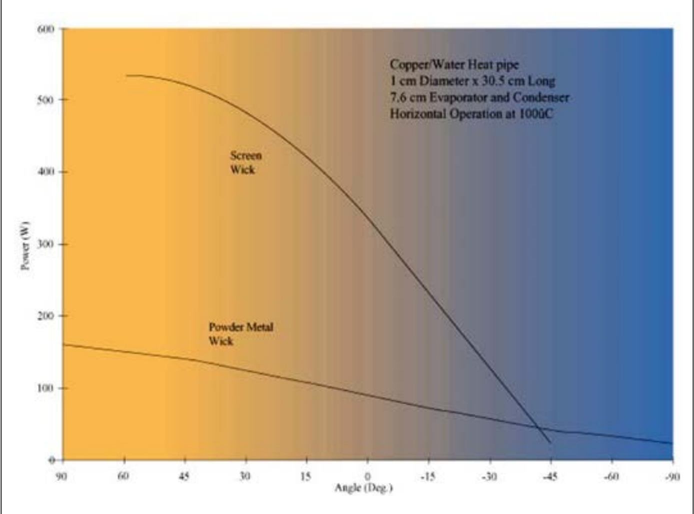

Figure 4: Capillary limits vs. operating angle

The two most important properties of a wick are the pore radius and the permeability. The pore radius determines the pumping pressure the wick can develop. The permeability determines the frictional losses of the fluid as it flows through the wick. There are several types of wick structures available including: grooves, screen, cables/fibers, and sintered powder metal. *Figure 5* shows several heat pipe wick structures.

It is important to select the proper wick structure for your application. The above list is in order of decreasing permeability and decreasing pore radius.

Grooved wicks have a large pore radius and a high permeability, as a result the pressure losses are low but the pumping head is also low. Grooved wicks can transfer high heat loads in a horizontal or gravity aided position, but cannot transfer large loads against gravity. The powder metal wicks on the opposite end of the list have small pore radii and relatively low permeability. Powder metal wicks are limited by pressure drops in the horizontal position but can transfer large loads against gravity.

EFFECTIVE HEAT PIPE THERMAL RESISTANCE

The other primary heat pipe design consideration is the effective heat pipe thermal resistance or overall heat pipe ΔT at a given design power. As the heat pipe is a two-phase heat transfer device, a constant effective thermal resistance value cannot be assigned. The effective thermal resistance is not constant but a function of a large number of variables, such as heat pipe geometry, evaporator length, condenser length, wick structure, and working fluid.

The total thermal resistance of a heat pipe is the sum of the resistances due to conduction through the wall, conduction through the wick, evaporation or boiling, axial vapor flow, condensation, and conduction losses back through the condenser section wick and wall.

Figure 6 shows a power versus ΔT curve for a typical copper/water heat pipe.

Figure 5: Wick structures

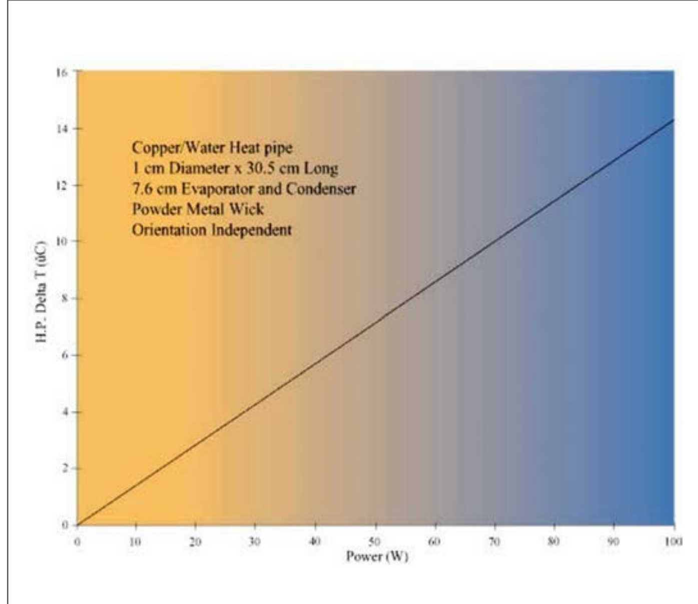

Figure 6: Predicted heat pipe Delta-T

The detailed thermal analysis of heat pipes is rather complex. There are, however, a few rules of thumb that can be used for first pass design considerations. A rough guide for a copper/water heat pipe with a powder metal wick structure is to use 0.2°C/W/cm^2 for thermal resistance at the evaporator and condenser, and 0.02°C/W/cm^2 for axial resistance.

The evaporator and condenser resistances are based on the outer surface area of the heat pipe. The axial resistance is based on the cross-sectional area of the vapor space. This design guide is only useful for powers at or below the design power for the given heat pipe.

For example, to calculate the effective thermal resistance for a 1.27 cm diameter copper/water heat pipe 30.5 cm long with a 1 cm diameter vapor space, the following assumptions are made. Assume the heat pipe is dissipating 75 watts with a 5 cm evaporator and a 5 cm condenser length. The evaporator heat flux (q) equals the power divided by the heat input area ($q = Q/A_{evap}$; $q = 3.8\ W/cm^2$). The axial heat flux equals the power divided by the cross sectional area of the vapor space ($q=Q/A_{vapor}$; $q = 95.5\ W/cm^2$).

The temperature gradient equals the heat flux times the thermal resistance.

$$\Delta T = q_{evap} * R_{evap} + q_{axial} * R_{axial} + q_{cond} * R_{cond}$$

$$\Delta T = 3.8\ W/cm^2 * 0.2°C/W/cm^2 + 95.5\ W/cm^2 * 0.02°C/W/cm^2 + 3.8\ W/cm^2 * 0.2°C/W/cm^2$$

$$\Delta T = 3.4°C$$

It is important to note that the equations given above for thermal performance are only rule of thumb guidelines. These guidelines should only be used to help determine if heat pipes will meet your cooling requirements, not as final design criteria. More detailed information on power limitations and predicted heat pipe thermal resistances are given in the heat pipe design books listed in the reference section.

HEAT PIPE ELECTRONIC COOLING APPLICATIONS

Perhaps the best way to demonstrate the heat pipes application to electronics cooling is to present a few of the more common examples. Currently, one of the highest volume applications for heat pipes is cooling the Pentium processors in notebook computers. Due to the limited space and power available in notebook computers, heat pipes are ideally suited for cooling the high-power chips.

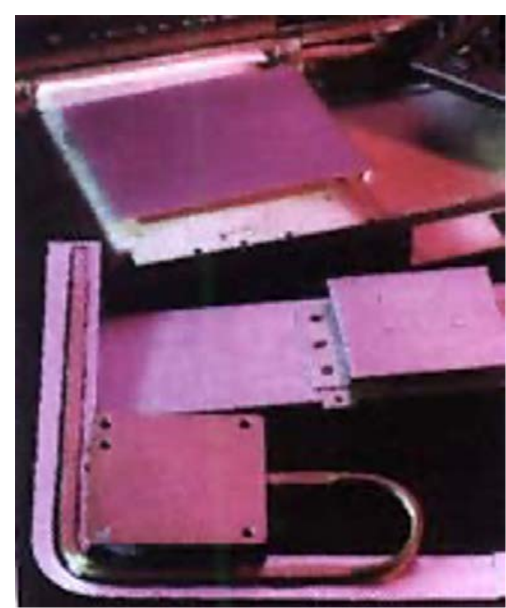

Figure 7: Typical notebook heat pipe heat sink

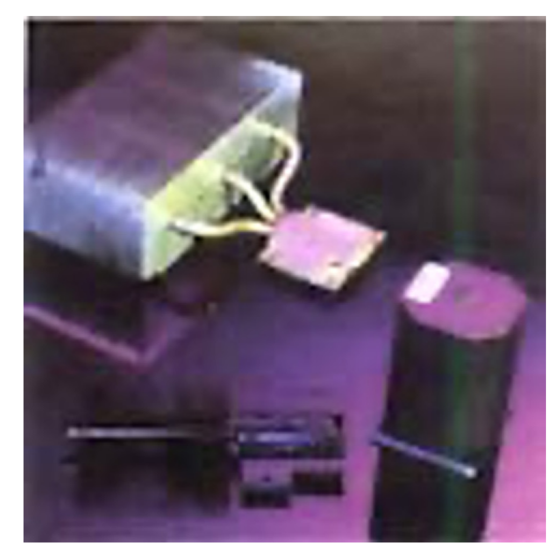

Figure 8: High-end CPU heat pipe heat sink

Figure 9: High-power IGBT heat pipe heat sink

Fan assisted heat sinks require electrical power and reduce battery life. Standard metallic heat sinks capable of dissipating the heat load are too large to be incorporated into the notebook package. Heat pipes, on the other hand, offer a high efficiency, passive, compact heat transfer solution. Three- or four-millimeter diameter heat pipes can effectively remove the high flux heat from the processor. The heat pipe spreads the heat load over a relatively large area heat sink, where the heat flux is so low that it can be effectively dissipated through the notebook case to ambient air. The heat sink can be the existing components of the notebook, from Electromagnetic Interference (EMI) shielding under the key pad

to metal structural components. Various configurations of notebook heat pipe heat sinks are shown in *Figure 7*.

In addition, other high-power electronics including silicon controlled rectifiers (SCRs), insulated gate bipolar transistors (IGBTs) and Thyristors, often utilize heat pipe heat sinks. Heat pipe heat sinks similar to the one shown in *Figure 9*, are capable of cooling several devices with total heat loads up to 5 kW. These heat sinks are also available in an electrically isolated versions where the fin stack can be at ground potential with the evaporator operating at the device potentials of up to 10 kV. Typical thermal resistances for the high-power heat sinks range from 0.05 to 0.1°C/W. Again, the resistance is predominately controlled by the available fin volume and air flow.

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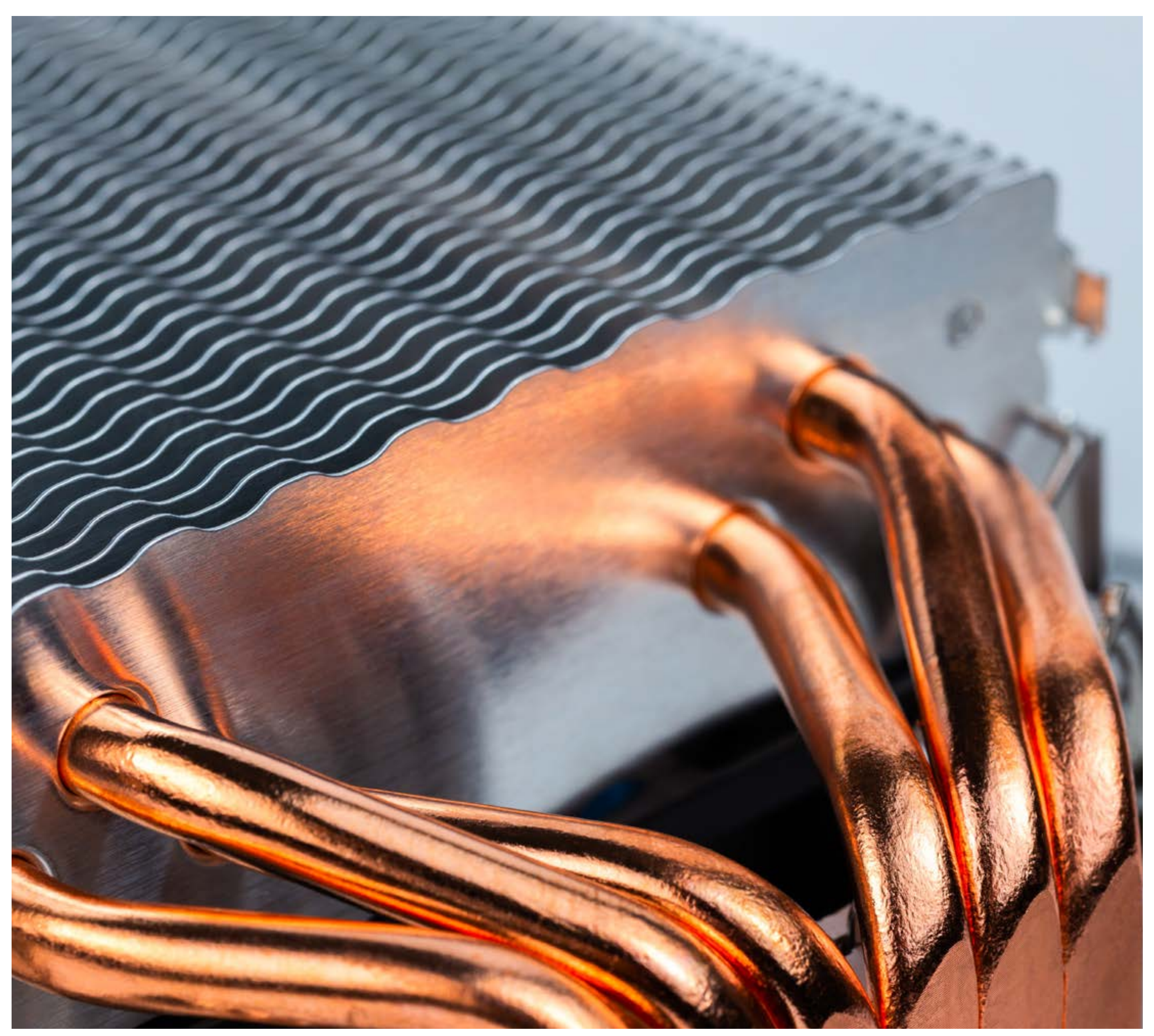

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Advanced Cooling for Power Electronics

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Sukhvinder S. Kang
*Aavid
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EDITOR'S COMMENTS:
This special edition spans 25 years of *Electronics Cooling*. Power electronics could not be omitted from the list. Over the years, power electronics became a hot topic (e.g. MOSFET, IGBT, data center, EV/HEV, etc.). This market is estimated at more than $10 billion with a compound annual growth rate of +4%. A diversity of cooling solutions can be used, but in the end, the final choice depends strongly on the application requirements. ***(Genevieve Martin)***

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

INTRODUCTION

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

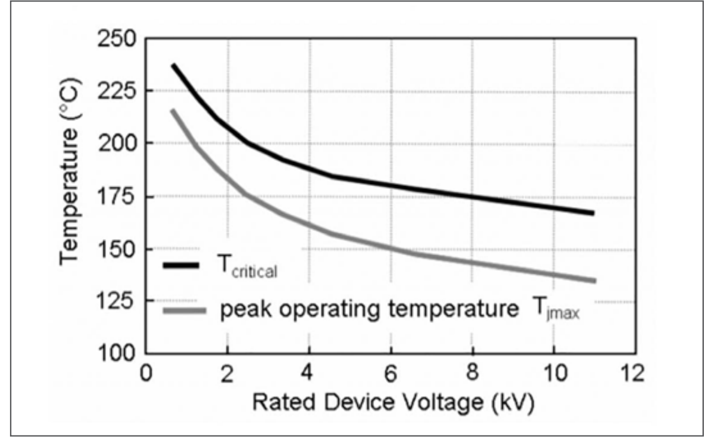

Figure 1: Critical thermal runaway temperature and estimated maximum safe operating temperature of Silicon devices [2]

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

Cooling solutions in use today include:

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

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

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

POWER MODULE PACKAGES

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

Figure 2: IGBT packages for single- and double-sided cooling [2]

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

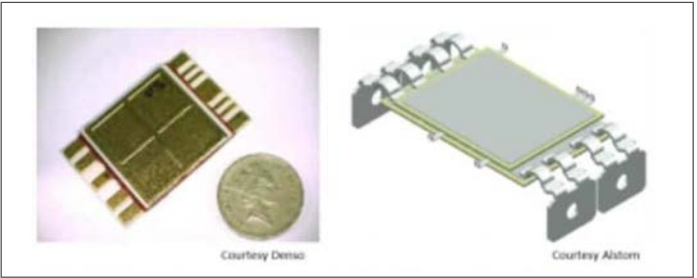

Figure 3: Custom power module designs for double sides cooling

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

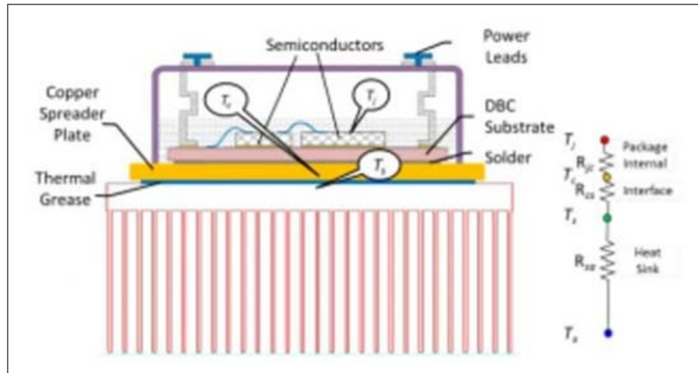

Figure 4: Schematic of standard IGBT package on a heat sink and key thermal resistances

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

$$R_{sa} = \frac{1}{\dot{m}c_p\left(1-e^{-\frac{hA}{\dot{m}c_p}}\right)}$$

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

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

$$\epsilon = \frac{T_o - T_i}{T_s - T_i}$$

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

FORCED AIR-COOLED HEAT SINKS

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

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

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

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

Table 1: Approximate thermal resistances in IGBT air-cooled assembly shown in Figure 5.

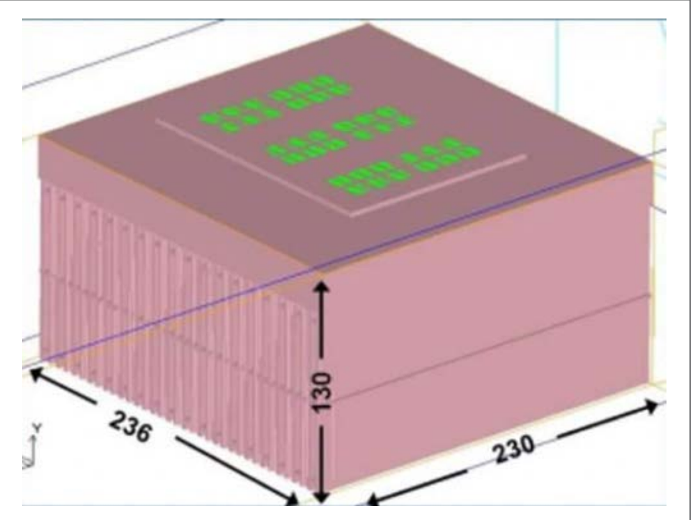

Figure 5: Illustration of IGBT module attached to an aluminum press-fin forced air-cooled heat sink

Figure 6: Air-cooled heat sink with copper base

Figure 7 shows an example of a heat sink with heat pipes embedded in the aluminum base. The copper water heat pipes use copper powder wick and are embedded in the base using Aavid's hi-contact technology. The heat pipes are oriented along the fin direction to even out the front to back temperature gradient due to heat up of the air. An ~60% reduction in the spreading resistance to 5°C/kW or a 14°C reduction in junction temperature can be achieved. The better heat spreading with heat pipes is achieved at a smaller heat sink mass compared to using a copper base. Recent advances using carbon nanotubes grown on copper mesh and powder wick enable heat fluxes in excess of 500 W/cm^2 [15] and reduction in thermal resistance. Such nanotechnology is expected to be available in commercial products based on market demands.

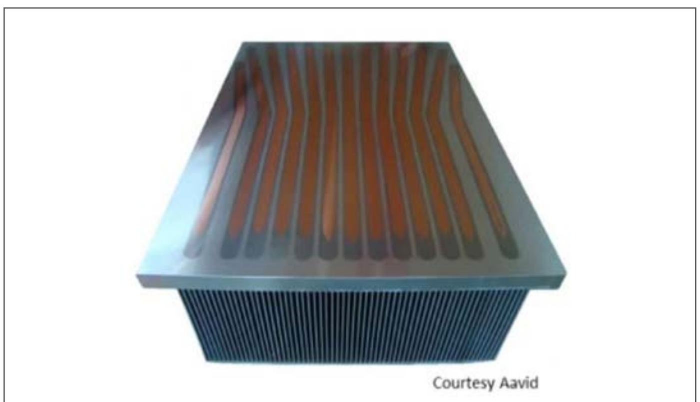

Figure 7: Air-cooled heat sink with heat pipes embedded in an aluminum base

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

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

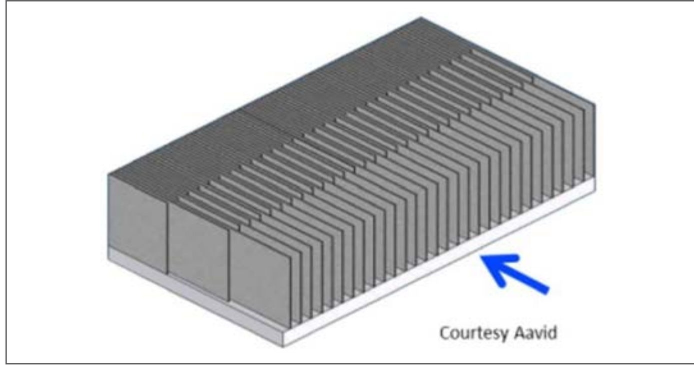

Figure 8: Advanced heat sink with increasing fin density along air flow direction

LOOP THERMOSIPHON AIR-COOLED HEAT SINKS

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

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

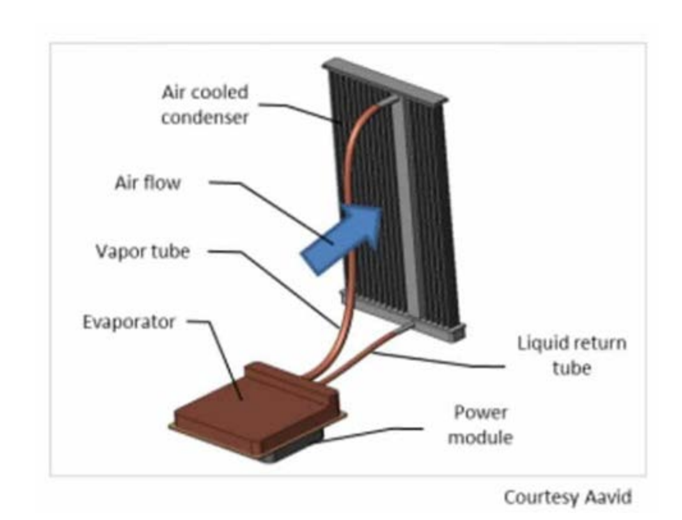

Figure 9: Air-cooled heat sink using loop thermosiphon

Thermal resistance of the thermosiphon heat sink can be determined by using the heat transfer coefficient of the boiling surface inside the evaporator per *Figure 10* and a condensation heat transfer coefficient of ~1 kW/m²-K inside the condenser tubing. Assuming a 122 x 162 mm IGBT dissipating 2 kW of heat, the heat flux on the evaporator wall is ~10 W/cm² and the boiling heat transfer coefficient is ~5.5 W/cm²-K. Including a 3 mm thick copper wall thickness, the evaporator thermal resistance is 1.3°C/kW. Assuming a 230 x 230 mm by 30 mm thick air-cooled condenser with 0.25 m³/s air flow rate (same as in *Table 1*), the estimated condenser thermal resistance is 9.1°C/kW so that the total thermal resistance R_{sa} is 10.4°C/kW, about a third of the 31°C/kW value for the press fin heat sink studied in *Table 1*. This huge improvement in thermal performance comes with the added benefits of lower mass, lower pressure drop, and an essentially isothermal heat sink base for good static current balance between parallel IGBT dies.

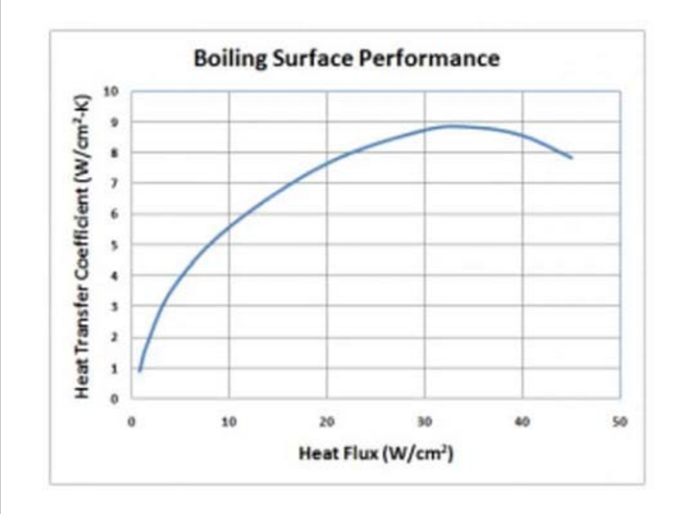

Figure 10: Heat transfer coefficient on thermosiphon evaporator boiling surface

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

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

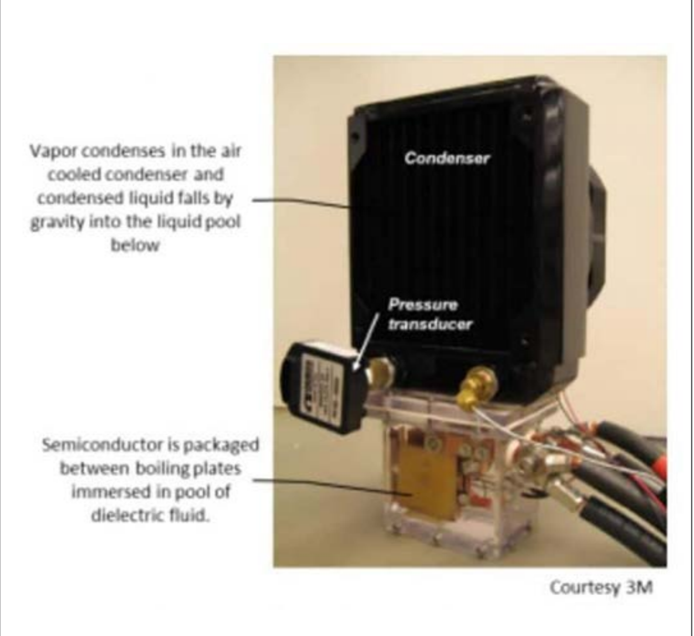

Figure 11: Thermosiphon air-cooled system with power electronics immersed in dielectric fluid. IGBT is clamped between enhanced boiling plates.

LIQUID COOLED COLD PLATES

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

- Tube type and fin type liquid cold plates to cool packages with DBC substrates with and without copper base plates.
- Liquid flow through fins formed directly on copper and AlSiC base plates.
- Single and double sided cooling using liquid jet impingement.
- Direct cooling of the base plate or DBC substrate using concepts such as the Danfoss "shower power" design using jet impingement or liquid flow through meandering channels.

- Direct double sided cooling of back to back modules with and without fins integrated with DBC substrates.
- Micro-channel coolers built into base plates or integrated with DBC substrate or into specially customized package designs.
- Stacked power modules and liquid cold plates using extruded channel type, folded fin type, and micro-channel type cold plates.

The paper [6] showed the great potential for achieving very high levels of cooling performance and system integration using liquid cold plates.

As noted earlier, *Equation (1)* guides us towards higher heat transfer coefficients using turbulent flow or small channel dimensions such as micro-channels to reduce thermal resistance. Two recently developed concepts that apply these principles effectively are presented next.

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

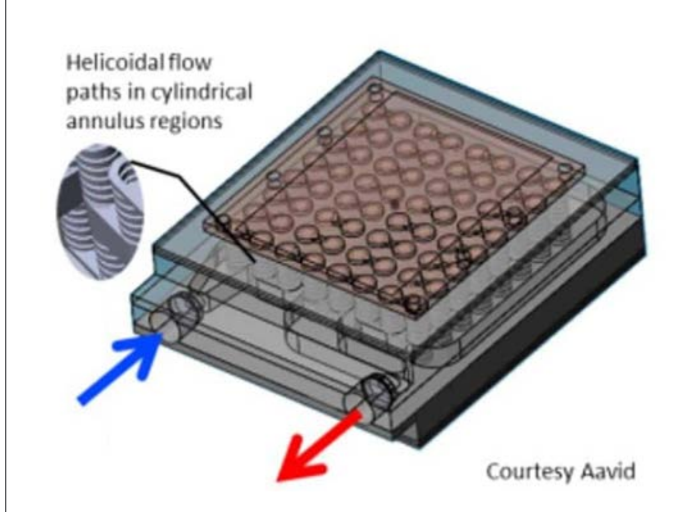

Figure 12: Vortex Liquid Cold Plate in a double sides cooling version

In the example considered to illustrate performance, an IGBT dissipating 1,600 W is attached to each side of the VLCP with thermal grease at the interface. The IGBT base plate size is 140 x 190 mm and uniform heat flux is assumed on the VLCP over the IGBT contact area. Cooling is provided by a water-glycol coolant with 50% glycol by volume flowing at 11 liters per minute (LPM) at a temperature of 80°C at the cold plate inlet. Under these conditions, the maximum temperature on the cold plate surface is 91.5°C, yielding a thermal resistance of R_{sf} = 3.6°C/kW. Using *Equation (2)*, the effectiveness is 47%. By comparison, the thermal resistances of suitable tube type and offset folded fin type liquid cold plates for the same application conditions were 19°C/kW and 5.6°C/ kW respectively. The pressure drop of the VLCP was 52 kPa versus 10 kPa for the tube type and offset folded fin type designs.

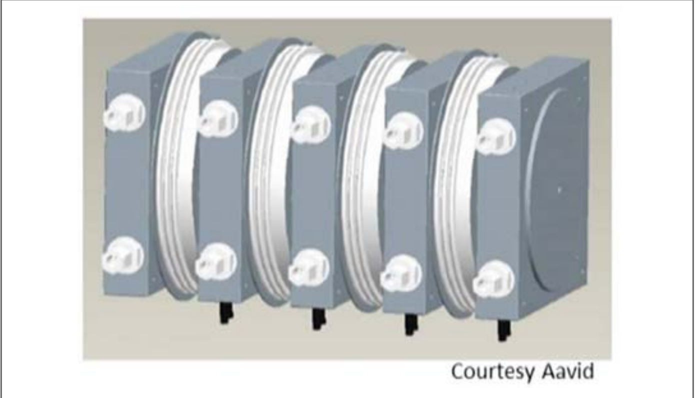

Figure 13: Stack of press pack diodes cooled using Vortex Liquid Cold Plates

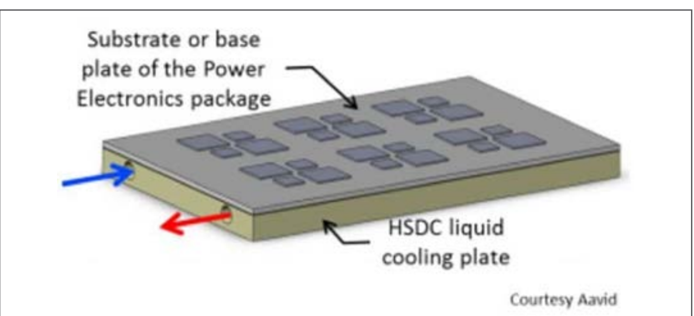

Figure 14: Liquid cooling of power electronics package using high shear direct contact concept

For liquid cooling systems where liquid filtration is possible, high heat transfer coefficients may be obtained using high shear flow in narrow channels. *Figure 14* shows a high shear direct contact (HSDC) cooling approach using a cooling plate that enables direct contact between the liquid and the surface to be cooled. An elastomeric O-ring (not shown) is provided between the cooling plate and the electronic package to seal the liquid flow volume. Liquid is distributed through a highly parallel manifold system and made to flow at a low velocity through narrow flow channels in direct contact with the surface to be cooled as shown in *Figure 15*. Ribs (not shown) are provided on the surface of the cooling plate that contact the cooled surface and set the height of the narrow flow channels.

The combination of low liquid velocity and small channel height insures laminar flow in the channels. By adjusting the spacing between the liquid supply and return channels in the manifold, the flow length of the cooling channels and liquid velocity can be set to achieve the desired pressure drop characteristics.

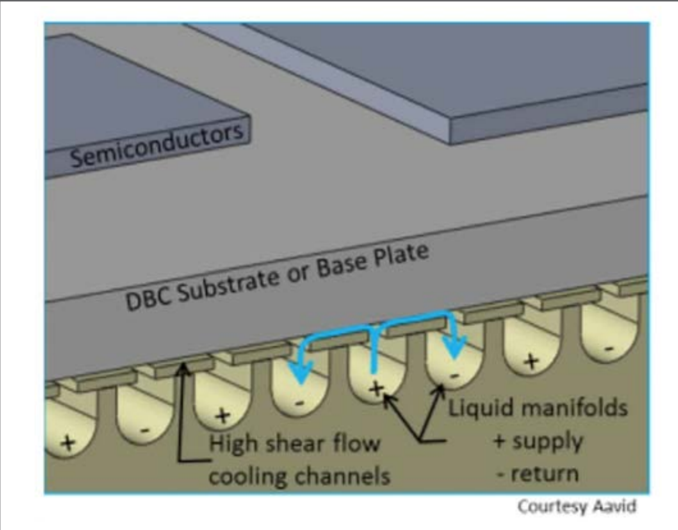

Figure 15: High Shear flow cooling channels and flow pattern in the HSDC liquid cooling plate

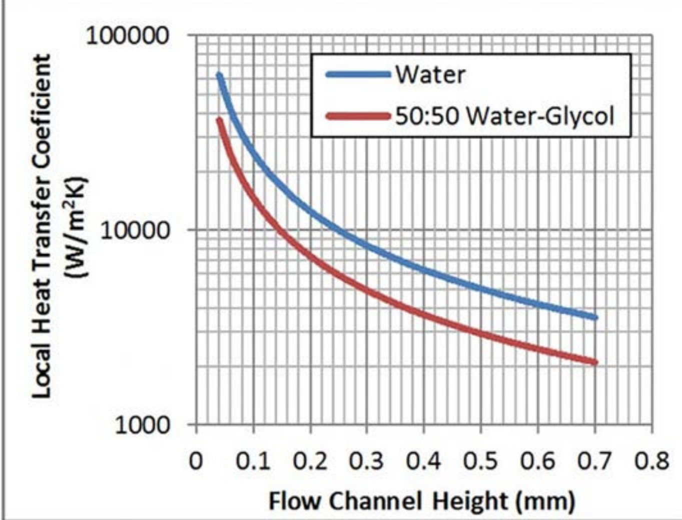

Figure 16: Local heat transfer coefficient on power module base plate at 80°C coolant temperature

It is obvious that the HSDC cooling plate may be designed to provide double sided cooling so that power electronics devices may contact the cooling plate on both the top and bottom surface. Furthermore, depending on the requirements of the application, the cooling plate may be made from non-conducting plastic or electrically conducting metal.

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

We will use the same application example as for the VLCP to illustrate the performance of the HSDC liquid cooling concept in a double-sided cooling configuration. The liquid contact area with the IGBT base plates on each side is 140 x190 mm and uniform heat flux is assumed over this area. Cooling is provided by a water-glycol coolant with 50% glycol by volume flowing at 11 LPM at a temperature of 80°C at the cooling plate inlet. We assume a flow channel height of 0.15 mm so that the local heat transfer coefficient is ~9,800 W/m^{2}K. From *Equation (1)*, the thermal resistance is ~2.8°C/kW so the maximum temperature rise on the cooled surface relative to the liquid inlet temperature will be ~9°C. Assuming each flow channel is 10 mm long, average liquid velocity in the channels will be ~0.5 m/s and the pressure drop will be ~3-5 kPa. This level of performance is very competitive with the other cold plate designs mentioned earlier but this concept does require coolant filtration to avoid clogging the very narrow channels. By reducing the flow channel height, the thermal resistance can be improved even further but at the expense of higher pressure drop.

CONCLUSIONS

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

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

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Nomenclature	
Q	Heat Load (W)
R_{ja}	Junction to air inlet thermal resistance [= $(T_j - T_a)/Q$ (°C/W)]
R_{sa}	Heat sink contact surface to air inlet thermal resistance [=$(T_s-T_a)/Q$ (°C/W)]
R_{sf}	Heat sink contact surface to fluid inlet thermal resistance [=$(T_s-T_f)/Q$ (°C/W)]
R_{cs}	Thermal interface resistance [=$(T_c-T_s)/Q$ (°C/W)]
T	Temperature (°C)

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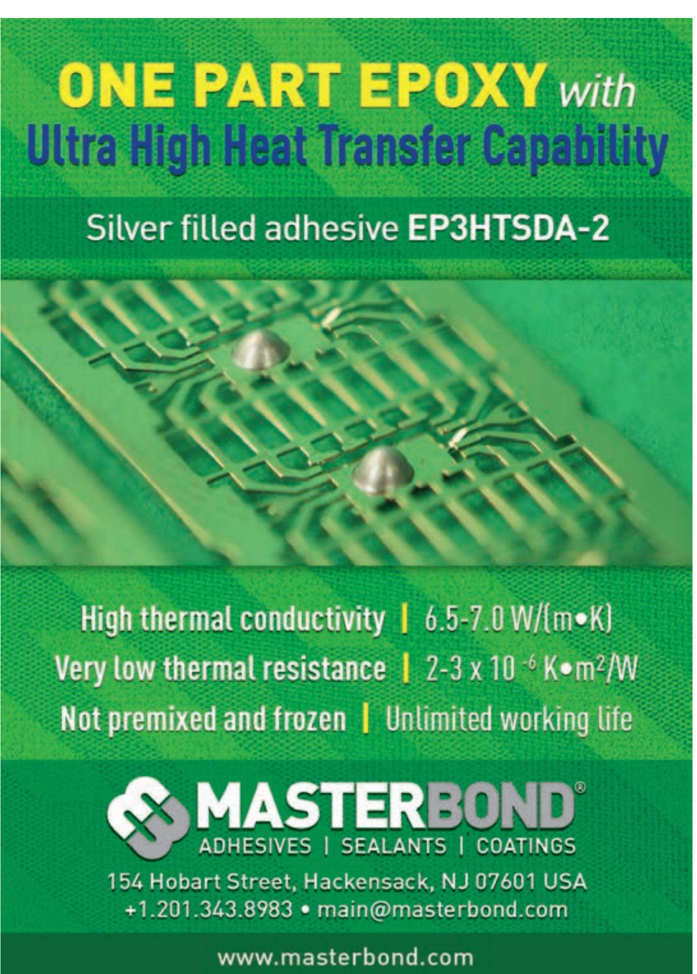

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