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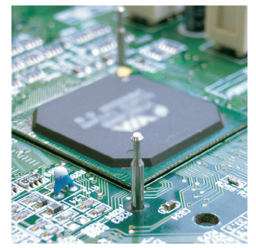

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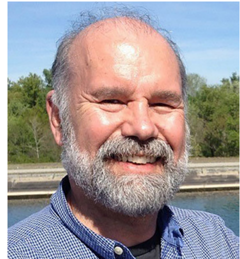
EDITORIAL

Data Center Power Trends – Where Do We Go from Here?



Bruce Guenin, PhD
Associate Technical Editor
Electronics Cooling, Summer 2018 Issue

Many of us in the thermal management industry have struggled with meeting the challenges in dealing with ever increasing power levels at the component level. We also remember warnings in 2000-2010 that the total energy consumed by data centers in the U.S. was increasing by 15% per year. Given our experience at the component level, we would have expected the data center energy consumption to continue at a 15% rate, or even greater.

Faced with this alarming trend, engineers began to worry not only about the power consumed by the computing and networking electronics, but also about the power used to operate the fans, air conditioners, and other cooling equipment. The term PUE (Power Usage Effectiveness), a measure of data center cooling efficiency was defined in 2006:

PUE = Total Facility Energy Consumed/Energy Powering the IT Equipment

For example, PUE = 1 if all the energy powers the IT equipment and PUE = 2 if the same amount of energy is devoted to cooling as to powering the IT equipment. PUE immediately began to be broadly applied to data centers. The ratings were, in general, closer to 2 than 1. Quantifying the problem led to broad thinking about how to reduce it.

Surprisingly, in the 2008 time frame, the energy consumption trend line bent so that the 15%/year slope was reduced to only 4%/year. This was certainly a puzzling development. Fortunately, there was a monumental study* by the U.S. Department of Energy that looked at all aspects of data center IT configurations, equipment, and cooling practices over the full range of data center sizes and provided an explanation.

The following table defines different data center size categories and indicates their average PUE value in 2014 and projected for 2020. It is noteworthy that only the hyperscale data centers have PUEs near 1. The high-end data centers have values around 1.7. All of the smaller ones have values in the range, 1.9 to 2.5. Furthermore, one expects only a small improvement in their PUEs by 2020.

The hyperscale data centers, built by cloud and internet giants such as Google, Facebook, Amazon, and Microsoft, have been designed from the ground up to have maximum cooling and power conversion efficiency and often incorporate selective liquid cooling.

Table -- Data Center PUE Values

Data Center Size Category	Typical Size		PUE Values	
			2014 (Actual)	2020 (Per Current Trends)
	(m^2)	(ft^2)		
Internal Server Closet	< 9.3	< 100	2.0	2.0
Internal Server Room	9.3-92.8	100-999	2.5	2.35
Localized Data Center	46.5-186	500-1,999	2.0	1.88
Mid-Tier Data Center	18.6-1,858	2,000-19,999	1.9	1.79
High-end Data Center	>1,858	>20,000	1.7	1.6
Hyperscale Data Center	Up to over 37,161	Up to over 400,000	1.2	1.13

The smaller data centers lack the economies of scale of their hyperscale cousins. They typically are entirely air cooled and often have suboptimum air flow conditions where there is poor separation between hot and cold air streams, leading to hot spots. Lacking the resources to better manage air flow, the common practice is to deal with the hotspots by overcooling the datacenter. With the emphasis on avoiding downtime rather than minimizing the power consumption devoted to cooling, these data centers tend to have higher PUEs.

The bending in the energy usage trendline was attributed by the report to the increased share of IT activity in the U.S. conducted by the hyperscale data centers with their very low PUEs. In spite of this trend, the cited report estimates that in 2020, 40% of the total energy will be consumed by data centers at the mid-tier size and smaller, with their lackluster PUEs of 2. Clearly, improving the energy efficiency of these small data centers would represent an opportunity for the thermal management community. Hence, we have decided to reprint two articles last published in our June, 2015 issue that are very relevant to this discussion.

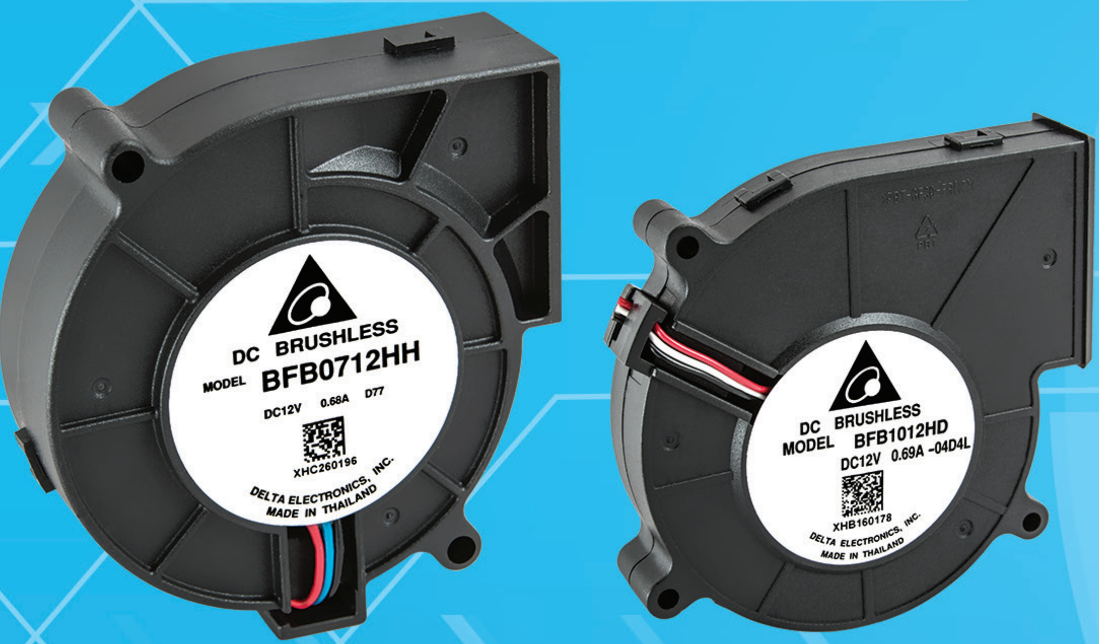
The first is, "A Simple Method to Understand Trade-Offs in Data Center Cooling." It provides a straightforward procedure for calculating the effect on cooling efficiency caused by effects such as trading off a higher cool air temperature against the need to run the fans at a higher flow rate.

The second article is, "Comparison of HPC/Telecom Data Center Cooling Methods by Operating and Capital Expense." It represents a very detailed, "bottom up" calculation of these expenses for air cooling, water cooling, and pumped refrigerant cooling. It synthesized inputs from the many members of the LinkedIn Liquid Cooling forum.

The issue is rounded out by the article, "Metal Foam Heat Exchangers" and an in-depth interview with members of a panel on the topic of a Neutral File Format for exchanging mechanical models used with thermal Computational Fluid Dynamics (CFD) simulation software in a portable, non-proprietary manner.

We hope you enjoy the issue!

*Arman Shehabi, et al. "United States Data Center Energy Usage Report," Lawrence Berkeley National Laboratory Document Number LBNL-1005775, June, 2016. [URL -- http://eta-publications.lbl.gov/sites/default/files/lbnl-1005775_v2.pdf]



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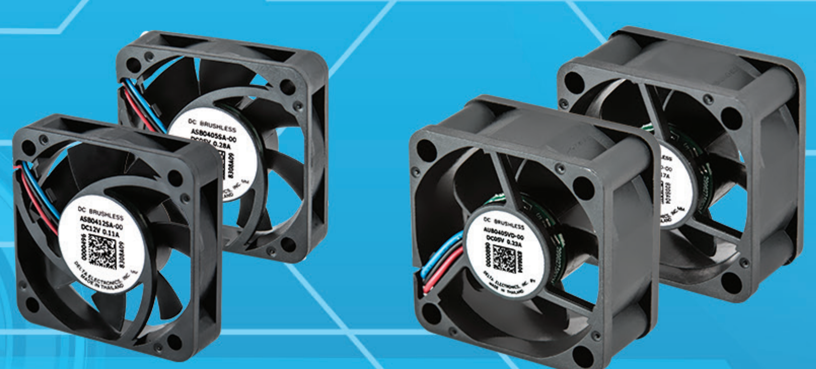

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A Simple Method to Understand Trade-Offs in Data Center Cooling

This Article Has Been Reprinted from the June, 2015, Issue

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Nomenclature	
COP	Coefficient of Performance
c_p	Specific heat
K_c	CRAC pressure coefficient
$\dot{m}$	Mass flow rate
N_c	Number of CRAC units
P	Power
T	Temperature
T_c	CRAC supply temperature
$\dot{Q}_c$	CRAC heat transfer
$\dot{V}_c$	CRAC air volume flow rate
Greek Symbols	
ΔT	Temperature rise
η	Efficiency
λ	Leakage air flow as a fraction of active cooling air
ρ	Density
ϕ	Hot air recirculation fraction
Superscripts	
i	Inlet
o	Outlet
*	Prescribed value
Subscripts	
c	CRAC
f	Fan
L	Leakage
r	Rack
t	Total cooling requirement
ref	Refrigeration
x	Data center exhaust

Cooling and thermal management are critical to data center reliability. Many organizations see cooling as a differentiating factor in the lifecycle cost of their data center. Recent industry guidelines [1] for data center cooling have suggested energy savings in air-cooled data centers by increasing the temperature of the cooling air. This increase enables two trends: reduced refrigeration power at higher refrigeration system evaporator temperatures and an increased number of hours available for using free-cooling. Free-cooling uses ambient air to provide cooling to the data center and reduce or eliminate the need for mechanical refrigeration. However, often overlooked is the significant power used in data centers for moving the cooling air. This article will use a simple thermodynamic and heat transfer analysis to highlight how trade-offs in Information Technology (IT) system power, computer room air conditioner (CRAC) fan power, and refrigeration power must be balanced to optimally operate the data center. Prior studies [2, 3, 4] have illustrated how simple models can be very effective at elucidating meaningful results.

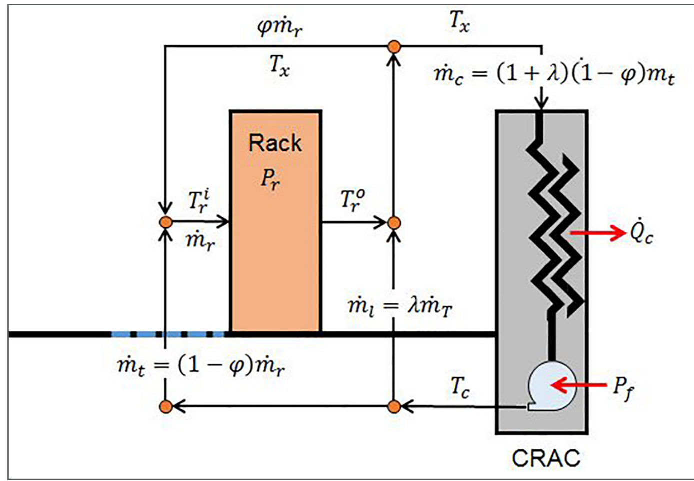


Figure 1 - Schematic of a Simple Model of an Air-Cooled Data Center

ESTIMATING THE AIR FLOW IN A DATA CENTER

Referring to *Figure 1*, which shows a diagram of a simple model of an air-cooled data center, the cold air emanating from the CRAC (at temperature T_C) is divided into an active cooling portion $\dot{m}_t$ and a leakage component $\lambda\dot{m}_t$, which bypasses the cold aisle and blends

with the IT rack's exhaust air. The leakage could represent air that escapes through cable cutouts or through gaps in the raised-floor. The IT rack consumes a given power P_r , which is a combination of the power required by the electronics and the power of the IT system's cooling fans. The cooling air that enters the IT rack is a combination of air supplied by the CRAC to the cold-aisle and a fraction, φ, of hot exhaust air that is recirculated back to the front on the IT equipment at the temperature of the CRAC return air, T_x.

A mass and energy balance at the inlet of the IT rack shows that the rack inlet temperature must satisfy,

$$T_r^i = (1-\varphi)T_c + \varphi T_x \ . \qquad (1)$$

The inlet air, after being heated by the electronics, exits the rack at a temperature,

$$T_r^o = T_r^i + \frac{P_r}{\dot{m}_r c_p} \ , \qquad (2)$$

where c_p is the specific heat of the air. The rack's exhaust air mixes with the leakage air to form the data center exhaust air. The data center exhaust temperature, which is both the recirculated air and the CRAC inlet air temperature, is given by an energy balance on the exhaust node as,

$$T_x = \frac{T_r^o + \lambda(1-\varphi)T_c}{1+\lambda(1-\varphi)} \ . \qquad (3)$$

Equation 1 can be combined with *Equation 3* to show that the data center's exhaust temperature must also satisfy,

$$T_x = \frac{T_r^o + \lambda T_r^i}{1+\lambda} \ . \qquad (4)$$

Lastly, an energy balance of the CRAC shows that the CRAC supply temperature must satisfy,

$$T_c = T_x - \frac{P_r}{(1+\lambda)(1-\varphi)c_p \dot{m}_r} \ . \qquad (5)$$

DATA CENTER COOLING POWER CONSUMPTION

The data center CRAC cooling power, P_c , is comprised of two components: a refrigeration component that is expected to be higher with a lower CRAC supply temperature and an air moving component that is expected to vary with the air flow rate as,

$$P_f = \frac{N_c K_c}{\eta_f}\left(\frac{\dot{V}_c}{N_c}\right)^3 = \alpha\left[\frac{(1+\lambda)(1-\varphi)}{N_c}\dot{m}_r\right]^3 \ , \qquad (6)$$

where, N_c is the number of identical CRAC units in the data center, K_c is the CRAC pressure coefficient, η_f is the overall fan efficiency, assumed to be driven by variable speed motors, and $\dot{V}_c$ is the total CRAC volume flow rate. The refrigeration power consumption can be computed from,

$$P_{ref} = \frac{\dot{Q}_c}{COP} = \frac{P_r + P_f}{COP} \ , \qquad (7)$$

where, $\dot{Q}_c$ is the CRAC's cooling requirement and COP is the refrigeration system's Coefficient of Performance. It can be obtained from an energy balance of the data center. Therefore, the total cooling can be computed as, $P_c = P_{ref} + P_f$.

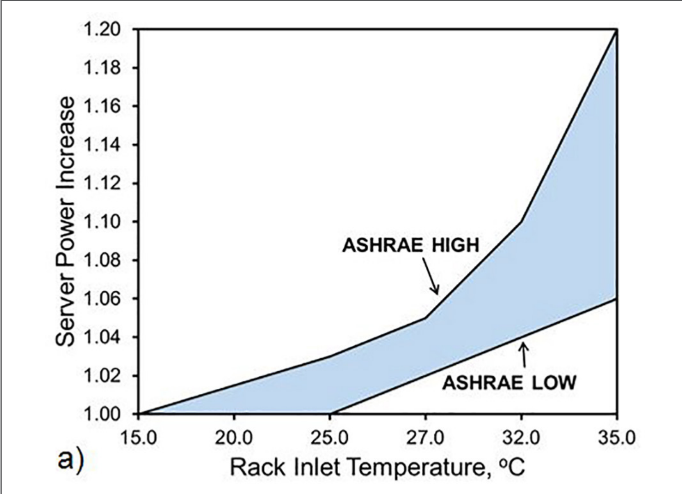
OPTIMIZED COOLING FOR AN ENCLOSED AISLE DATA CENTER

As an example, let's assume that the data center has implemented cold-aisle containment [2, 5]. Theoretically, in an enclosed aisle data center, the air provided through the perforated tiles must equal the air required by the racks (i.e., $\phi = 0.0$). The above set of equations can be used to parametrically study the optimal operating condition, given values of λ, K_c, η_f ,the variations of rack power and rack flow rate, and the variation in the CRAC COP. *Table 1* provides the parameters used in the analysis.

The variation in rack flow rate and power is not straightforward and the design varies from manufacturer to manufacturer. This example uses the relationships provided by ASHRAE [1], which have been reproduced in *Figure 2*, which shows that, as a function of inlet temperature, the required power and air flow rate can increase by 20% and 250%, respectively, as the inlet temperature is increased from 15 to 35°C. The figures also highlight the variation in IT equipment design by expressing the relationships as a band, with high performance and highly utilized servers as the upper limit (ASHRAE HIGH) and lower-utilized, lightly-configured servers as the lower limit (ASHRAE LOW). To use these figures, we will prescribe the rack power, P_r^* , at 15°C and the server's temperature rise ΔT_r^* at 15°C. The required rack flow rate at 15°C can then be computed from *Equation 2*. With the IT equipment characteristics defined at 15°C, the trends in *Figure 2* can be used to find the power and flow requirements at higher inlet temperatures. For the CRAC performance, we will use the simple relationship described in [6], which expresses the CRAC COP as a monotonically increasing function of the CRAC supply temperature.

Table 1: Parameters Used in Optimized Cooling Example Analysis for an Enclosed Aisle Data Center

Parameter	Value
P_r^*(kW)	1000.0
ΔT_r^*(°C)	15.0
λ	0.20
K_c(Pa/(m^6/s^2))	20.0
N_c	12
η_f	0.65
ρ(kg/m^3)	1.225
c_p(kj/kg-K)	1.012



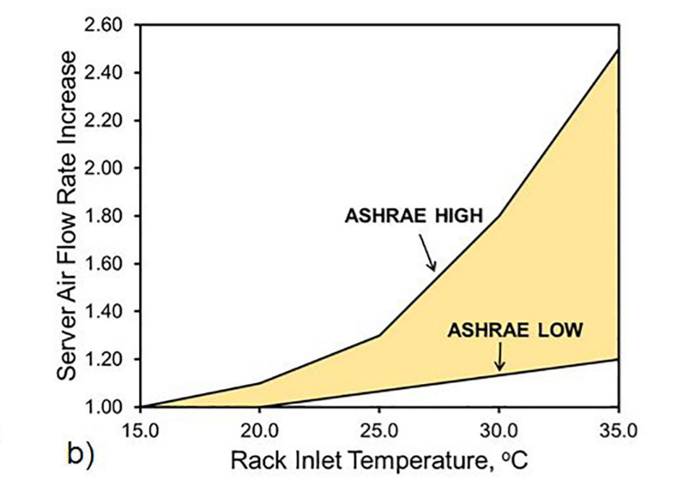


Figure 2: a) ASHRAE Volume Server Power Increase with Inlet Temperature and b) ASHRAE Volume Server Flow Rate Increase with Inlet Temperature. Reproduced from [1].

Using *Equations 1 – 7,* the relationships in *Figure 2*, and the parameters defined in *Table 1*, the data center's cooling power requirement can be computed as a function of rack inlet temperature. *Figure 3* shows the cooling power as the rack inlet temperature is varied from 15 to 30°C. It shows that the optimum operating point (i.e., the point with lowest cooling power consumption) occurs around 23°C. The figure shows the trade-off that data center operators should consider. Increasing the inlet temperature increases the amount of power and airflow required by the IT equipment and thus the amount of air that must be delivered by the CRACs. Even though the increased temperature enables a reduction in refrigeration power, this reduction is not sufficient to overcome the increased power requirement of the CRAC fans. Exclusive focus on reducing refrigeration power consumption without regard to the power consumed in moving the cooling air would lead to suboptimum and possibly misleading results.

Figure 3 highlights the trend in cooling power for ASHRAE high-power and highly-utilized IT equipment. Using the trends from ASHRAE in *Figure 2*, *Figure 4* shows how the design of the server's cooling algorithm, utilization, and power consumption can impact the data center cooling optimization. High-performance and/or highly-utilized servers exhibit a clear data center minimum power consumption point; whereas, lower-utilized and/or lightly-configured servers, which have lower air flow requirements, exhibit their minimum data center power consumption point at higher inlet temperatures. Interestingly, for this class of servers, the reduction in power consumption beyond 24°C is relatively small, compared to the savings going from 15°C to 24°C. Since uptime and reliability are paramount to data center design, a designer may consider operating at a lower temperature and sacrifice small energy savings, to provide more resiliency in the event of a cooling failure.

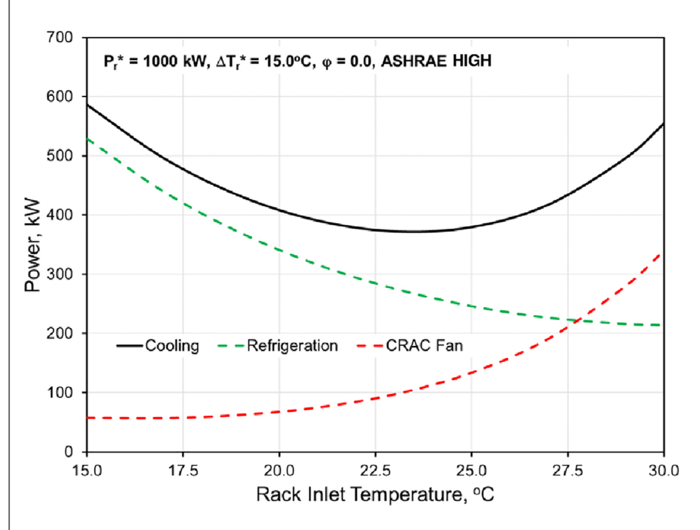


Figure 3: Predicted Data Center Cooling Power Consumption Breakdown based on Equations 1 – 7 and Table 1 Parameters.

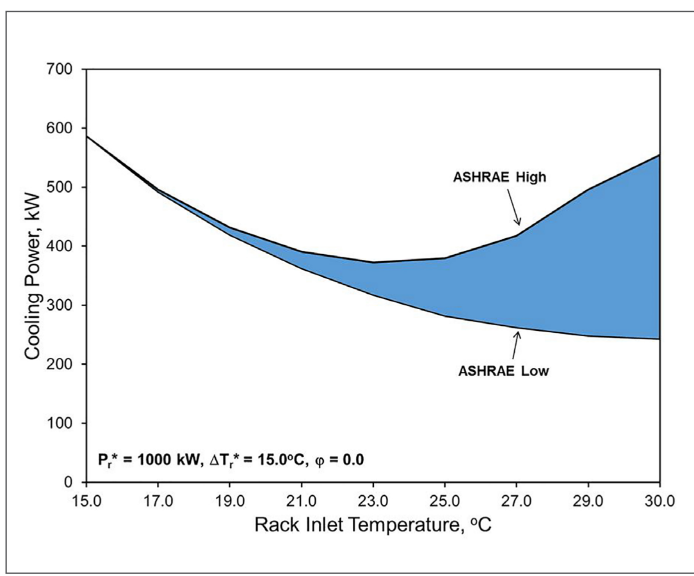


Figure 4: Predicted Volume Server Impact on Data Center Cooling Optimization based on Equations 1 – 7 and Table 1 Parameters.

CONCLUSIONS

This article shows how a simple thermodynamic and heat transfer model can be useful in determining the most efficient manner for cooling IT servers in a data center. It highlights the trade-off between refrigeration power and air moving device power, that data center operators need to consider in order to optimize their data center's cooling system. Hopefully, this article offers the reader a methodology that can be considered at the early design stage of any thermal-fluid system in order to fully understand the problem at hand, before considering more detailed and time-consuming techniques.

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The Discussion Over a Neutral File Format for Thermal CFD Heats Up

Host:
- **Jean-Jacques (JJ) DeLisle** | Executive Editor of *Electronics Cooling*

Panel:
- **David Ochoa** | Senior Thermal Engineer for Intel Corporation
- **John Parry** | Electronics Industry Manager for Mentor's Mechanical Analysis Division
- **Chris Aldham** | Software Development Manager for Future Facilities
- **Lawrence Der** | Senior Product Marketing Manager at Cadence Design Systems

David Ochoa

David Ochoa received a BS and MS in Mechanical Engineering from the University of California, Davis. He has been working in the electronics cooling field since 2003 and is currently employed by Intel Corporation's Data Center Group as a Senior Thermal Engineer. David has used multiple commercial CFD packages and has developed internal software analysis tools for solutions specific to Intel's thermal challenges. It has been in this work that David realized the need for a standard format for CFD and is committed to its success for the benefit of the electronics cooling industry.

John Parry

A chemical engineer by training, John Parry earned a PhD in reactor design before getting involved with computational fluid dynamics (CFD) more than 25 years ago. He joined Flomerics Ltd. when it opened its doors in 1989 and continued there when it became the Mechanical Analysis Division of Mentor Graphics, now a Siemens business. As the division's Electronics Industry Manager, John has coordinated EC-funded projects and managed Knowledge Transfer Programs and strategic internal projects as well as overseeing the technical integration of Mentor's Microelectronics Research and Development (MicReD) business into the division.

John's technical contributions to the discipline of electronics cooling include developing a host of compact thermal models for fans, heat sinks, chip packages and LEDs. He has published several papers in these areas, and also has expertise in Design of Experiments and optimization. Today Dr. Parry is also responsible for the Mechanical Analysis Division's research activities and its Higher Education Program. He serves on the JEDEC JC15 Thermal Standards Committee and on various conference committees, and was General Chair of the SEMI-THERM 21 conference.

Chris Aldham

Dr. Chris Aldham has worked in computational fluid dynamics (CFD) for over 37 years (starting with PHOENICS at CHAM with Prof. Brian Spalding) and for more than 26 years in the field of electronics cooling. After 16 years at Flomerics Chris joined Future Facilities in 2008 as a Product Manager responsible for 6SigmaET – electronics cooling simulation software. Chris is currently Future Facilities Software Development Manager responsible for a team of 15 coders, testers and technical writers.

Lawrence Der

Lawrence Der is a Senior Product Marketing Manager at Cadence Design Systems working on the Sigrity Analysis products. Prior to Cadence, Lawrence held various senior product management and design engineering roles in consumer electronics and fabless IC companies. He received his Ph.D. in electrical engineering from the University of California, Los Angeles and M.S. and B.S. degrees in electrical engineering from the University of California, Davis.

INTRODUCTION

A discussion in the electronics thermal engineering community has been brewing for the past several years, and it's only getting more heated. The root of the discourse is a desire from thermal engineers to have a Neutral File Format for mechanical models used with thermal Computational Fluid Dynamics (CFD) simulation software. Currently, there are several thermal CFD simulation softwares, and each operates with its own proprietary, or customized, mechanical file formats. Hence, if an engineer is working with multiple softwares, or a vendor is providing thermal CFD models for their customers, a new mechanical model must be created for each specific simulation software. This can be a taxing and error prone process, and result in additional man-hours of work to create the new models and confirm their accuracy.

Though this has been a long standing issue, the topic has boiled to the surface at the recent Semi-Therm 2018 in San Jose, California. Through a series of presentations and discussions, representative from electronics component vendors and thermal CFD software companies have renewed this discussion as an active initiative.

While the discussion is still in its early phase, Electronics Cooling is presenting a panel interview with leading thermal engineers and experts with the goal of summarizing and illuminating various points of view on the topic.

PANEL INTERVIEW

• **JJ –** Many other industries operate with several proprietary and non-proprietary file formats for modeling and simulation, so why is a Neutral File Format for thermal CFD a topic for discussion? Why now?

• **Ochoa –** The effectiveness of these types of formats is already realized in many other disciplines, mechanical models have STP and PCB software uses Gerber to name a few. Computational fluid dynamics is somewhat of a niche engineering specialty, and while the usefulness is recognized, the initiative hasn't previously found a voice to coordinate a standard. Recently, thermal engineers from Intel and Motorola found the occasion to start the discussion and get the ball rolling.

Additionally, there is a general growing trend that simulation tools are evolving to meet the need of co-simulation capability. For example, when a mechanical analysis of a structure is exposed to temperature and flow variations, CFD becomes essential for a meaningful solution. The prospect of having a non-proprietary format will allow a simple path to co-simulation without being tied to a specific format and adds even more momentum to this initiative.

• **Parry –** Broadly as I see it, some of the companies at the top of the electronics supply chain want to reduce their work by only supplying thermal models of their parts in a single neutral format rather than providing models for specific simulation tools like FloTHERM. One reason this might be happening now may be due to the increase we are seeing in thermal design activity as power levels continue to rise. The increased use of stacked die and packages has increased power densities at the package level back up to levels that are difficult to cool.

A neutral file format has been proposed as a way forward by two major organizations in the industry. This has been seized upon as a business opportunity by the smallest of the CFD vendors supplying the electronics market, who have offered to open up their internal file format. Other companies, including Mentor, are currently evaluating the risks vs benefits and trying to understand fully the functional and non-functional requirements and how a comprehensive solution really benefits end users.

• **Aldham** – Thermal simulation of electronics has been dominated by 2 major players for the last 20+ years. Most companies selected one tool or the other and some used both. As the ability to handle more detailed and complex models has evolved due to increased computer power the need to make extreme approximations has reduced. One area where this is manifest is a need for better component models – which has led to a dependence on component manufacturers to supply accurate models of their components. As the file formats for each software is different these must be supplied in software specific forms.

Over the years a number of other companies have entered the electronics cooling simulation arena and these struggle to get established as the 'supply chain' isn't necessarily able to supply models in an appropriate format. And where these are available it's additional work for the manufacturers.

Even when only supplying models in a limited number of formats, the effort required runs into a significant number of man hours for each manufacturer. The supply chain is much bigger and more interconnected than it was 20 years ago. The market needs data that is readable extensible and 'light-touch' to avoid errors.

Furthermore, newer software often has a different feature set or improved performance which could attract users to switch from their old software to a new company's software. Unfortunately, users often find it difficult as there is little or no compatibility between models created in the different software.

• **Der** – From an EDA perspective, we are seeing a growing number of our power integrity customers request model files for thermal components as the electronics industry continues to develop products that are faster, smaller, smarter, and more complicated. Unfortunately, many of these model files are only available in proprietary formats, forcing the component suppliers or our customers to make new models that work with our simulators. As we know, this is an inefficient and error-prone process. A common Neutral File Format can solve this problem by enabling thermal and electrical component suppliers to develop a single common file that works with simulation tools from all vendors supporting the format.

• **JJ** – What are the potential benefits and who stands to gain from the development and standardization of a neutral file format for thermal CFD?

• **Ochoa** – CFD software can be complex, requiring training and years of experience to use effectively. It is natural that an engineer is predisposed to using one CFD package over another and would have difficulty sharing simulations with others. This is also a common problem for component suppliers, such as Intel, that must provide multiple types of models to enable customers to design systems with adequate cooling. Without a common format, the only solution has been to bear the cost of supporting multiple file types with the risk of duplication error.

Besides saving resources by eliminating the need to create duplicate models, CFD packages may have strengths and weaknesses and enabling the ability to transfer models can allow an engineer to make use of the best simulation strengths by any package.

Another benefit is allowing interdisciplinary advancements in modeling capability. Having a generic format allows data transfer into other engineering tools where co-simulation is possible. A great example of this is Cadence® has planned to support this format with Sigrity™ PowerDC™ to share information between simulation toolsets.

• **Parry** – Theoretically the benefits are that thermal models can be shared between tools. A lesson from the past is that in practice this often means agreement on what is the lowest common denominator that can be easily supported by a neutral file format, and that can be quite inadequate for users. On the one hand, only a small subset of a model can be exported that way, and on the other, if the model is built such that it only uses those object types, etc. that can be written in that format, doing so would be very inefficient from the user's perspective. It is not clear to me that end users and others in the electronics supply chain naturally benefit from such an approach.

• **Aldham** – Everyone involved in electronics cooling can benefit. Component manufacturers (and others in the supply chain – for example fans, disk drives, power supplies etc.) would only have to provide models in one format for all end users regardless of the software they use. Currently they can spend many hundreds, if not thousands, of man-hours per year preparing models in different formats and reducing that to a single format would provide significant savings. End users would benefit from having a free choice of software to use – not being bound to only the limited choice supported by the manufacturers. This could encourage the establishment of a more robust supply chain for electric thermal simulation which, I think, would lead to more accurate and reliable simulations throughout the industry.

• **Der** – The benefits are time savings and a potential reduction in model errors for our customers, the product development engineers. Ultimately, this can benefit the end customer with better designed products that meet costs because the engineers will have more time to focus on the product design.

• **JJ** – What information would a Neutral File Format for thermal CFD need to contain for it to be viable?

• **Ochoa** – The primary goal is to enable passing mechanical and thermal attributes, such as geometry, material properties, boundary conditions, flow sources ect. Because these tools are complex there won't be a 1:1 transfer of all information.

The responsible engineer will always need to review the model, optimize it within the software limitations, and ensure the validity of the gridding scheme.

• **Parry** – What is really required is an agreement to standardize on a mechanical CAD format for the mechanical geometry. Since today's electronics are so tightly integrated with the mechanical design of products much of the geometry is only available as CAD parts. On the EDA side, tools like Xpedition from Mentor are going fully 3D, so again the geometry is essentially 3D CAD. Rather than start with what can be done simply and quickly, which has the drawback of not being extensible to cover the range and richness of information that end users really need, it would be better to define up front what is needed to cover everything that end users build in simulation tools today, then work out how to deliver that in a set of agreed stages.

• **Aldham** – The proposal from Intel had some information about the sort of items that would need to be address by the neutral file format. Obviously, it must contain the geometry of the various items, material names & properties and particular thermal attributes such as power dissipation.

• **Der** – Please see the recommendations from Intel.

• **JJ** – On the list of essential information for a Neutral File Format would be mechanical model information. There are several methods of describing physical geometries, which method is most viable for a neutral file format, and why?

• **Ochoa** – Solving the governing equations for heat and fluid flow require simplifying geometry wherever possible. The geometry needs to be broken down into primitive shapes such as cubes, cylinders, cones, and prisms. While software vendors differ on how they decompose complex geometry imported by CAD tools, there is a base level of simple shapes that are common and easily described via coordinate and size tags within the XML format.

• **Parry** – My feeling is that it should be some widely-used format for 3D solid geometry that supports parametric feature data, so that the model can be evolved. Parasolid might be an option. Simple 3D solid geometry like IGES, and worse still surface definitions like stereolithography (STL) are quite inadequate for simulation tools. Another tack would be to just cover geometry that

can be defined by parameters, as we have with our SmartPartsTM, so covering fans, heatsinks, package compact thermal models, and simple building blocks like cuboids. That approach may be adequate for companies who are building boxes, like servers and switches to go into a data center, however many of our customers are designing products that need mechanical CAD parts such as in a phone for example, where the enclosure is itself an important part of the thermal management solution.

• **Aldham** – The geometric capabilities of Flotherm from Mentor are relatively crude and because of its large user base the neutral file format would need to be quite low level. As they don't deal with CAD shapes directly in the software but just some primitives such as cuboids and prisms the neutral file format would have to be based around those shapes. It would be nice to have some more complex geometry handling such as STL or STEP but most electronics cooling software doesn't really use these formats directly in their models.

• **JJ** – It has been proposed that a potential format that is commonly used for complex data, such as that for mechanical models for thermal CFD, could .XML. What are the benefits of using .XML as a file format, what other options are there, and are there any drawbacks to these formats?

• **Ochoa** – The primary benefit of XML is that major CFD vendors already use it in an encrypted form. XML is an effective way to share data in a structured format without limitations on tagged attributes, meaning it can easily be extended as need arises in the future. Since software vendors already favor and use this format, we aren't requiring a major development effort on their part to adopt and standardize a non-proprietary version.

Most tools can already import geometry using an existing standards (STL, IGS, STP for example), however thermal attributes and common items such as fans, vents, and PCB's will not transfer. The drawback is then that an engineer must spend considerable time reapplying these features.

• **Parry** – From a software engineering perspective, XML is a format that is used extensively in the industry and forms the basis for many proprietary formats in use today. Writing parsers based on a given XML schema is a straightforward software engineering task and on-going maintenance, extension, and updates can be handled with relative ease. From a user's perspective XML really allows the author to give meaning to the data. For example, a CAD part that is in fact a cuboid, or a cylinder, or other simple geometric shape could be flagged as such, and more advanced objects such as a heatsink SmartPart could be identified. However, the real value of XML is that it allows additional attributes to be added; for example, a material with the relevant material thermal data, surface information such as roughness, emissivity, and color. So whatever parametric solid geometry format is chosen, XML can be used to add the additional information that a CFD tool needs.

• **Aldham** – XML seems to be a good choice as it's quite a flexible format and most software vendors will be familiar with it and probably already use it in some way. The XML tag names are readable and convey the meaning of the data. The information structure is easily discerned by both humans and computers as each XML tag immediately precedes the associated data.

The data structure follows a noticeable and useful pattern, making it easy to manipulate and exchange the data. Also, I think it would be possible to extend the format with extra tags which specific software could use or ignore as appropriate.

• **JJ** – Do any other nonproprietary modeling standards exist, for example with mechanical CAD, or other prior work, that could be a starting point for the development of a neutral file format for thermal CFD?

• **Ochoa** – CGNS stands for CFD General Notation System and was developed in the aerospace industry for similar purposes in the early 1990's. While I'm not familiar with the current usage, this standard undertakes a much larger scope than is needed here. For example, CGNS passes details such as solution data, specific thermal and flow models, convergence history and grid definitions. Since modern CFD tools used in electronics cooling have grown in capability, they can vary greatly on these items and it is not feasible to pass such detail.

My own prior work with this software tells me we need to keep it as simple as possible and easily extendible as CFD evolves.

• **Parry** – Yes, historical examples exist from the MCAD world, and each vendor will have their own internal format in which they hold the data. These have typically not been developed with simulation in mind, and so they do not form a good starting point. One example from the EDA world is IDF. IDF 2 and 3 were published and widely adopted. The inadequacies of these formats were identified, and a new highly extended format published as IDF 4, but it was not widely adopted, so it is yet another legacy format that is no longer actively being worked on.

One format that is certainly worth consideration is PLMXML (https://www.plm.automation.siemens.com/en/products/open/plmxml/), which is emerging as an open format for facilitating product lifecycle interoperability.

• **Aldham** – There are CAD standards such as Step and IGES but in general these do not hold all the information necessary for a thermal (CFD) solution. In addition most current electronics cooling software would need extensions to read or write these formats (ie. do not have a CAD kernel internally) as well as extensions to the format to hold the additional thermal data (heat dissipated, fan curves, etc,) necessary for a full model description.

I know Ansys and Mentor have developed a shared format to exchange some model data but this is not available or publicized

generally I think. This could perhaps be a good starting point for the neutral file.

The neutral file will have to operate at quite low level and reading/writing the file will require significant interpretation of the objects into each software model. This means that some aspects of the model in a specific software might be lost but overall the basic features should be maintained. (In 6SigmaET we have lot of specific objects which do not exist in the same form in Flotherm or Icepak (for example, PCB, PCB layer, Component, Resistor, etc.). However, each of these can be represented in s simple basic form that Flotherm/Icepak could understand (i.e. cuboid, cylinder etc.) and the neutral file would have to operate (at least initially) at this level.)

• **JJ** – What is a viable process for development such a file format? Who would need to be involved, and who would responsible for the development of the format?

• **Ochoa** – To be successful, I think we need to involve a large representation of the electronics cooling industry. Intel for one, we've solicited input from our industry colleagues and have honed in on a list of attributes that need to be included, the most important of which is that it remain nonproprietary.

We would like the leading CFD vendors to take it from here and offer an XML schema that satisfies the industry listed needs. We have already had one vendor volunteer an XML schema, and expect others to follow.

• **Parry** – Although slow, a standardization body would be the right forum for such an endeavor. The reason for the process being slow is that the issues are complex.

• **Aldham** – I suspect we need Intel and perhaps Motorola to take the lead from an industry 'neutral file user' point of view to ensure that the developments meets their needs in the marketplace with respect to the supply chain. A small group of software companies, Mentor, Ansys, Future Facilities, Cradle, could work with Intel/Motorola to ensure the format could be read written by their software (and help with establishing a common set of data that can be exchanged).

• **JJ** – What are the foreseeable barriers in establishing a standardized neutral file format for thermal CFD?

• **Ochoa** – Getting CFD vendors on board is the critical barrier. They may perceive an open format as a competitive threat or opportunity for other companies to enter this space. I hope they weigh this with the opportunities to develop co-simulation advantages and to further differentiate their products.

• **Parry** – Time and the willingness of all parties to work together. We have been active in the JEDEC JC15 committee since the mid-1990s and pioneered the development of guideline on the creation of 2-Resistor and DELPHI compact thermal models of packages.

• **Aldham** – In some respects the neutral file format could be seen as breaking down the virtual duopoly that Mentor and Ansys and making it easier for others to compete for thermal simulation users and therefore they may not be keen.

It does appear that, at least initially, the neutral file format can only contain the lowest common denominator data across the various software products. This may limit the successful model creation for any exchanged data – potentially undermining the automation and intelligence that can exist when models are built at a higher level.

• **Der** – Standardizing a format can take a long time and can also require considerable negotiation from all parties to reach consensus on the actual format.

• **JJ** – Large standards bodies that often handle such matters exist, such as JEDEC. Why would such a standard need to be developed outside of JEDEC to meet current needs.

• **Ochoa** – We see JEDEC as a necessity for long term success since the specified format needs to be hosted and updated in the future. For speed and simplicity, I believe we can successfully develop rev 1 of this standard separately, but eventually need a standards body to maintain.

• **Parry** – JEDEC may not be the best forum for this, but it should not be done outside of a standards body, because many companies would only be willing to supply their data in an additional file format if that is standardized. One of the reasons that JEDEC has been of value to the industry is that end users can demand metrics and other thermal data that the JEDEC JC15 committee has standardized on over the years. If a metric is a JEDEC standard it is harder for a vendor to refuse to measure and provide that information.

• **Aldham** – The main reason is one of speed. Traditionally bodies such as Jedec have moved at glacial pace. I feel it would be better to get 'something' established and then hand it over to a standard body.

• **Der** – Standards bodies such as JEDEC are essential in creating and maintaining open standards to ensure product interoperability for high-volume markets. In this case, I believe the thermal community should first establish an open neutral model file format to replace proprietary model formats. Once a format is available and adopted by enough companies, it can become the default public domain format. This neutral file format could later be adopted/modified by a standards body if needed.

• **JJ** – What progress has been made so far with developing such a format? What are the next steps?

• **Ochoa** – The first step has been to align key industry participants, get inputs and ensure that we were meeting critical needs. This has been a self-propagating effort since most of the industry

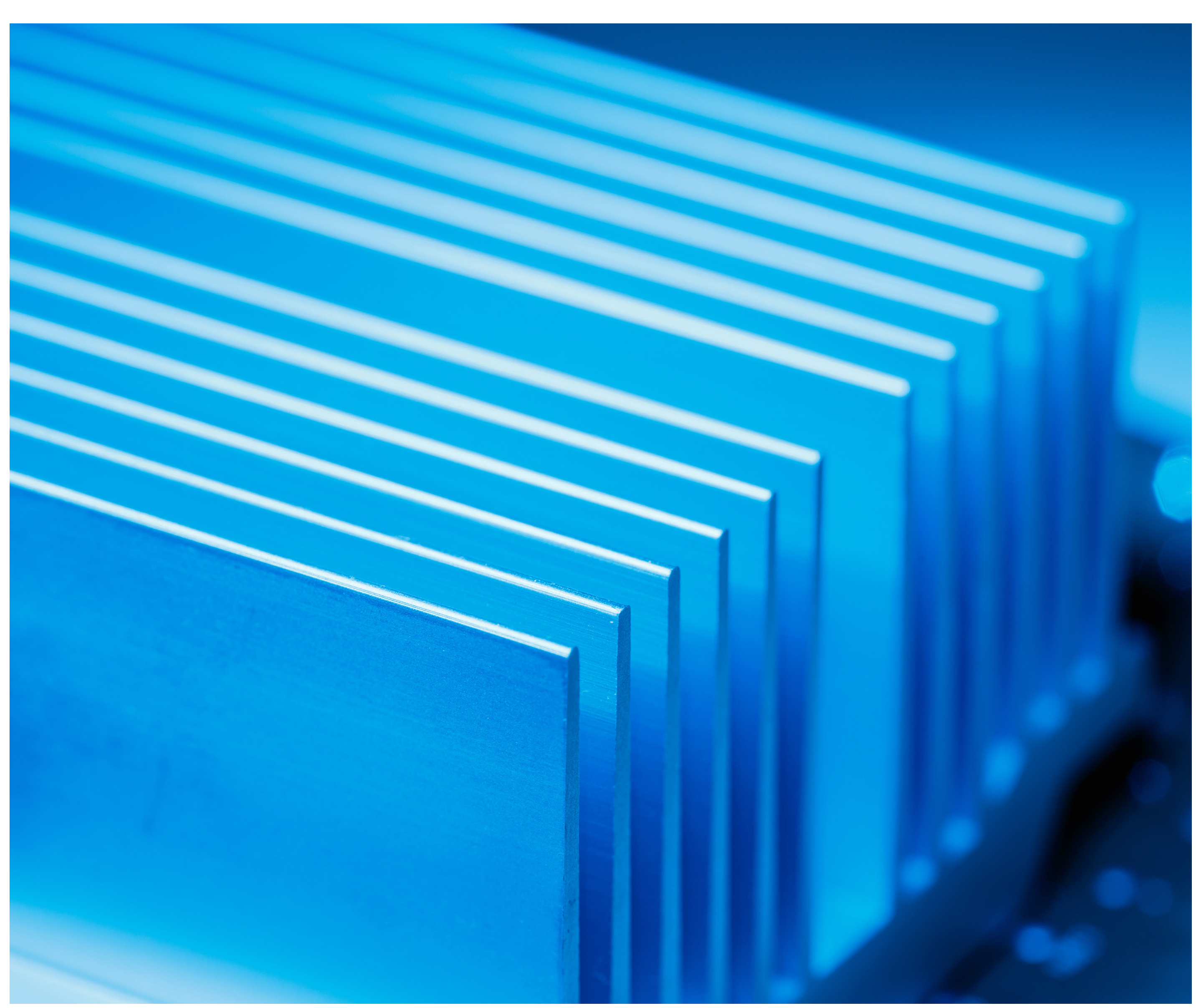
immediately recognizes the benefit and offers their support. Next, we are actively soliciting vendor feedback and will collectively endorse the XML schema that can be implemented easily while meeting industry requirements. Progress is happening as one vendor has already submitted an XML schema and other vendors are ready to adopt it or integrate into their software for co-simulation purposes. The ball is certainly rolling!

• **Parry** – Actually, our work in the JEDEC JC15 committee has borne fruit. The JEDEC JC11 committee has just published a file format that includes a neutral format for 2-Resistor and DELPHI compact thermal models of packages, so in that context, a standard format already exists.

• **Aldham** – As far as I know a few preliminary meetings and discussions have taken place. The 'public' meeting at Semi-Therm was the first time a larger group met to discuss the issue where Intel presented the requirement and motivation for the format. Future facilities offered to open their data format for use as an exchange format but in it's complete form that could contain a lot of data that was of limited use to other software. It could be interpreted or ignored as appropriate.

The next step must be to establish the group of Users (Intel/Motorola) and Software vendors to start discussing the finer points of the format. If the Mentor/Ansys format was available as a starting point then obviously that could accelerate the process and initial discussions could focus on any shortcomings/restrictions and how extensions could be made and managed.

• **Der** – Future Facilities has released their format as an initial candidate for a neutral file format. Intel has requested feedback and also opened an invitation to computational fluid dynamics software vendors to collaborate on a format.



Metal Foam Heat Exchangers

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Reticulated Metal Foams (RMF) are a cost-effective and ultra-high-performance thermal management materials that can be integrated with electronic devices and modules. RMFs are compatible with DI water, inert fluoro-carbons, jet fuel, inert gases.

STRUCTURE OF RMF

There are quite a few ways to fabricate RMF [1], however the investment casting method of manufacturing produces the most desirable material properties. In the as-fabricated state, the isotropic RMF consists of randomly oriented polygon shaped cells that can be approximated as dodecahedron, *Figure 1* [2,3,4]. Notice that the cross sections of roughly 2 mm long solid ligaments are mostly triangular. The geometry of RMF cell structure and the high purity and ductility of its metal produce the most desirable characteristics for heat exchanger (HX) applications. The physical dimensions of its structure, as shown below, does not allow boundary layers to grow and introduce enhanced mixing through eddies and turbulence. These features result in a high local film coefficient. RMFs metal foams commonly have 5, 10, 20 and 40 pores per inch (PPI) configuration and 4-13% theoretical density fabricated with, 6061 Al, C10100 Cu or Ag, among others. The important parameters of the RMFs are; thermal conductivity, heat transfer surface area, high mechanical ductility and compliance.

Thermal Conductivity:

The foam manufacturing process preserves the high purity of the material in the RMF. The thermal conductivity of 6061 Al and C10100 Cu, most common RMF materials, are about 170 W/m-K and 390 W/m-K, respectively. The effective bulk thermal conductivity however, depends on the porosity of the foam RMF's effective bulk conductivity (ke) may be estimated by *Equation (1)* [2].

$$k_e = \lambda . k_b . \rho \qquad (1)$$

Where:
λ, the proportionality constant $\lambda = 0.346$
k_b, the thermal conductivity of the base material
ρ, the porosity (relative density) of as foamed RMF ~ 8%

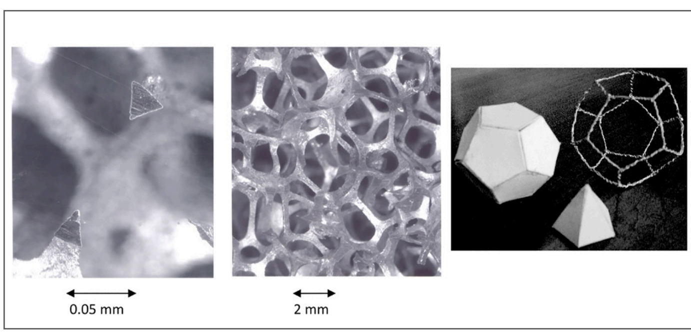

Figure 1. 40 pores per inch (PPI) 6101 Al based metal foam consisting of nodes and ligaments forming a space filling network of dodecahedrons with 12 pentagon shaped facets.

The effective bulk conductivity of 8% dense 6061 Al RMF is about 4.7 W/m-K. Due to their high ductility, RMFs can undergo significant inelastic and elastic buckling deformations without failure

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of the ligaments, resulting in an increase in the relative density of the foam structure up to 50%. Since the thermal conductivity is a vector quantity, its value will be a function of not just the amount of compression (as it is for the effective surface area), but also of the direction of compression. The effective thermal conductivity of 6061 Al based foams biaxially increases to ~ 30 W/m-K when unidirectionally compressed in X direction to 50% relative density in the (YZ) plane where ligaments are aligned in Y and Z directions. As the cost depends on the volume as well, this feature allows optimizing both the thermal performance and the cost of RMF HX effectively and simultaneously [2].

Surface Area Density:
One of the most important features of the RMFs is their extremely high and scalable surface area density (ρ_s) compare to those of brazed or extruded fins and fin pins. ρ_s is directly related to the extended surface area for improved convective heat transfer. The ρs of RMF was characterized using experimental measurements, by multipoint Brunauer, Emmett and Teller (BET) method by adsorption of krypton gas at 77.4 K, and the modelling studies by authors.

The results of these studies showed that ρs of 40 PPI RMF at as fabricated 6% and compressed 50% state are about 15.5 cm^2/cm^3 (40 in^2/in^3) and 138 cm^2/cm^3 (350 in^2/in^3), respectively [2,3].

Thermal Interfaces and Convective Film Coefficients:
An RMF based compact heat exchanger can be integrated to the sources of heat generation via solder bonding. Integration eliminates the highly resistive thermal interfaces of soft materials such as thermal pads, pastes or thermal epoxies commonly used to couple discrete devices, Hybrid Multi-Chip Modules (HMCM) of photonic and electronic devices to cold plates. An RMF may be brazed to low expansion skin layers and function as a constraining double-sided core heat exchanger (HX) for printed wiring boards (PWBs).

The RMF structure has a very high effective compliance, [2] that allows metallurgical bonding to the foam by soldering or brazing to low CTE materials (metalized ceramic plates, low expansion composites, Mo, and CuMoCu, among others). Since the CTE mismatch related thermal stresses and deformations are limited, the reliability of the integral heat exchanger, and the thermal base is not compromised, as verified by several hundreds of thermal cycles [5].

The mostly triangular cross section and only a couple of millimeter long ligament geometry of RMF offers significant advantages in convective cooling. It scales down the thickness of boundary layers, thereby generating vortexes and, inducing early transition to turbulent flow, and similarly delays or eliminates the transition from nucleate boiling to film boiling. The net outcome is enhanced heat transfer due to high local film coefficients.

Fabrication of RMF Heat Exchangers:
Method of manufacturing of RMS heat exchangers depends on the material and the design. Al RMF based HXs can be fabricated with vacuum or dip brazing. Fully enclosed HX/cold plate (CP) configurations require vacuum brazing using solid braze pre-forms. HXs with exposed RMF can be fabricated by either dip brazing or vacuum brazing. The open cell structure of RMF allows cleaning of any residual salts left over from the dip brazing bath. The advantage of vacuum brazing however becomes apparent for manufacturing in larger quantities. Use of a vacuum furnace may accommodate hundreds of units in a single batch operation at a lower cost per unit.

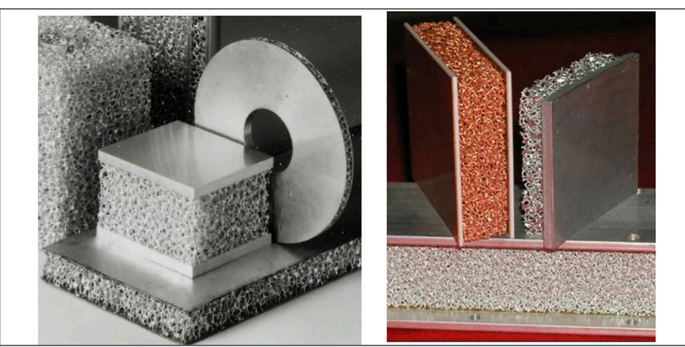

Figure 2. Precursors of Al and Cu RMF vacuum brazed HXs and CPs

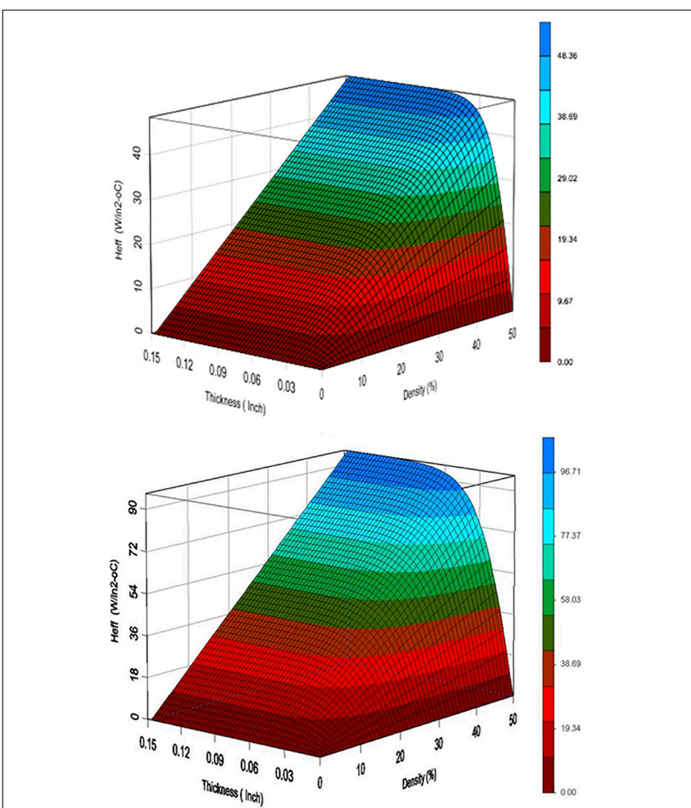

Figure 3. Effective h of 40 PPI 6061 Al (top) and Cu (bottom) RMF HXs with DI water at 63.1 cm^3 /sec (1 GPM). The thickness range for both the Al and Cu foams is 0-38 mm. The ranges of effective heat transfer coefficient for the Al and Cu foams are 0 to 7.5 (W/cm^2 °C) and 0 to 15 (W/cm^2 °C), respectively.

The fabrication of Cu foam-based heat exchangers where Cu foam is bonded to a Cu plate of enclosed housing are fabricated with inert-atmosphere, high-temperature brazing or vacuum brazing furnaces with suitable Cu-Ag solid braze preforms. Solder pastes may be used to fabricate CPs in inert-atmosphere furnaces where RMF is exposed. *Figure 2* shows aluminum foam-based articles made by vacuum brazing [6].

Thermal performance of RMF Heat Exchangers:
The major factors scaling the thermal performance of RMF HX are:

- Thermal conductivity of the base material (Al, Cu, Ag or others).
- Pore size measured as PPI, the linear density of pores per inch (5-40 ppi).
- Relative density (5% to ~50%)
- The thickness (similar to fin efficiency)
- Thermo-physical properties of the coolant

The recommended liquid coolants are distilled (DI)water, ethylene glycol, jet fuel, lubricating motor oils, Castrol, inert fluoro-carbons etc. Distilled water at 1 GPM flow rate was used as the coolant for generating the thermal performance surfaces shown in *Figure 3* [2].

Experimental Study of Thermal Performance:
The test module was fabricated by inert atmosphere brazing a 2.54cm x 2.54cm x 0.635cm (1.00" x 1.00" x 0.250") Cu block to the center of a 5.08cm x 5.08cm x 0.318cm (2.00" x 2.00" x 0.125" thick) Cu. A Plexiglas housing of the same cavity depth was fabricated out of a 0.635cm (0.250") thick Plexiglass sheet and screwed to the Al and Cu plates with cork gasket. A 2.54cm x 2.54cm (1.00"x1.00") resistor was eutectic Sn/Pb soldered on the center of the Cu Plate in inert gas environment. The flow rate and the inlet temperature of the inlet DI water were kept constant using a recirculating chiller. The volume flow rate, inlet and exit temperature of the coolant were monitored via flow meters and thermocouples. The surface temperature of the cold plate was estimated by the temperature measurements with thermocouples on the resistor. Pictures and drawing of the cold plate are depicted in *Figures 4* and *5* respectively. The experimental setup and the test cold plate are shown in *Figure 6*.

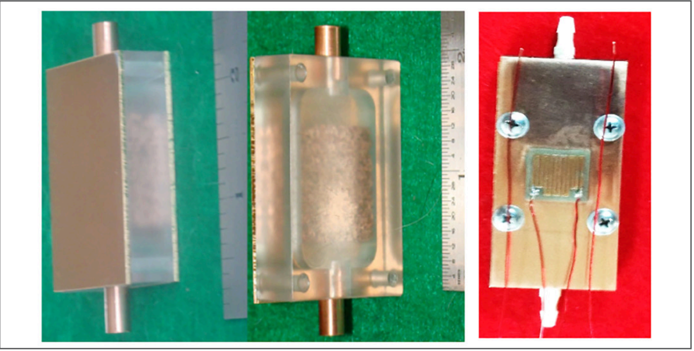

Figure 4. Double and single sided see through functional cold plates (left), top view of the single sided test unit (right).

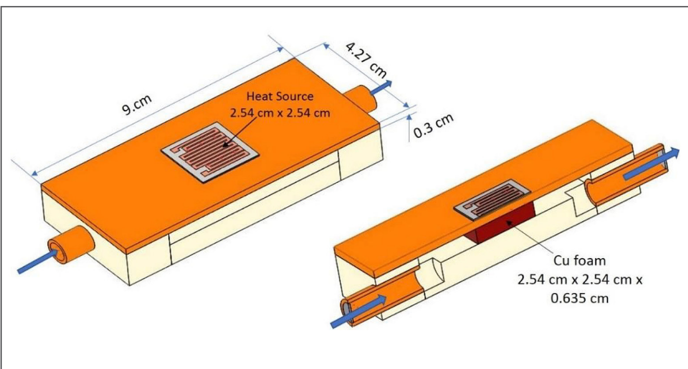

Figure 5. Figure 5. Drawing of the Cu RMF Cold Plate

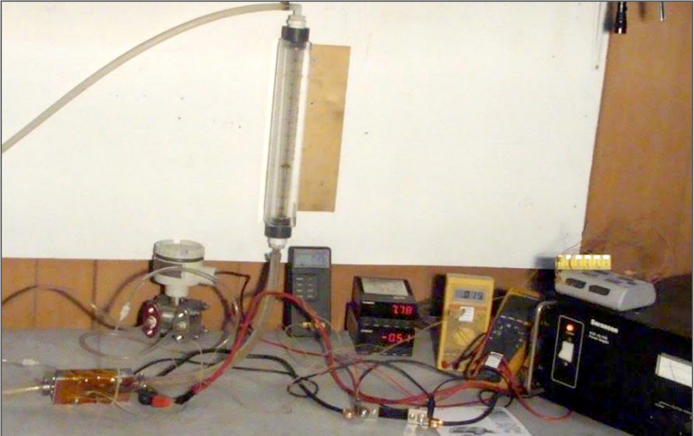

Figure 6. Unit under flow and thermal resistance tests

Calculation of Effective Film Coefficient

The estimated local film coefficient of 1 $W/cm^{2o}C$ for 63 cm^3/sec (1 GPM) flow rate was used in generating the thermal performance surfaces shown in *Figure 3* [2].

The vertical axis is the effective flat plate film coefficient whereas the X and Y axis of the surface plot show the thickness and the density of RMF, respectively. The effective film coefficient is proportional to the density. The thickness of the RMF has a linear relationship with the effective film coefficient at low thickness which asymptotically approaches its saturation value with increased thickness, similar to that of the fin efficiency.

The average effective film coefficient may be estimated from the measured power input to the resistant heater, and the difference between the average temperatures of the coolant and the resistor as was done in authors laboratory. The results of such tests agree with the results of the calculations as presented in *Figure 3*.

Thermal Performance Comparisons:

The cold plate performance requirement for a high-power electronics device is defined by the thermal resistance. In the early stages of design, the feasibility of a given cold plate technology may be assessed by its thermal resistance (R_{th}) which is commonly calculated by *Equation (2).*

$$R_{th} = \frac{\Delta T}{P} \quad (2)$$

Where;

P: Power dissipated by the device

ΔT: Temperature difference between the maximum allowable surface temperature of the CP and the exit temperature of the coolant.

ΔT may be calculated by *Equation (3)* where ρ_w=1000 (kg/m^3), V= 6.3*10^{-5} m^3/sec (1.0 GPM), C_{PW}=4184 (J/kg-^{o}C), T_{in}=21°C, and the maximum allowed surface temperature of the cooled electronic device T_{Max}=60°C.

$$\Delta T = T_{Max} - (T_{in} + 0.5.P / \rho_W.\breve{V}.C_{PW}) \quad (3)$$

$$\Delta T = 41.9 \quad {}^oC$$

R_{th} and Pressure Drop Measurements and Calculations:

R_{th} was calculated using the above procedure. Specifically, the value of R_{th} at V= 6.3*10^{-5} m^3/sec (1.0 GPM) is equal to 0.042^{o}C/W. The thermal resistance of the high performance micro channel type Cu cold plates, among others, is about 0.05^{o}C/W under the same conditions. Graphs plotting values of R_{th} and pressure drop versus flow rate, based on measurements and CFD analyses, are presented in *Figure 7*.

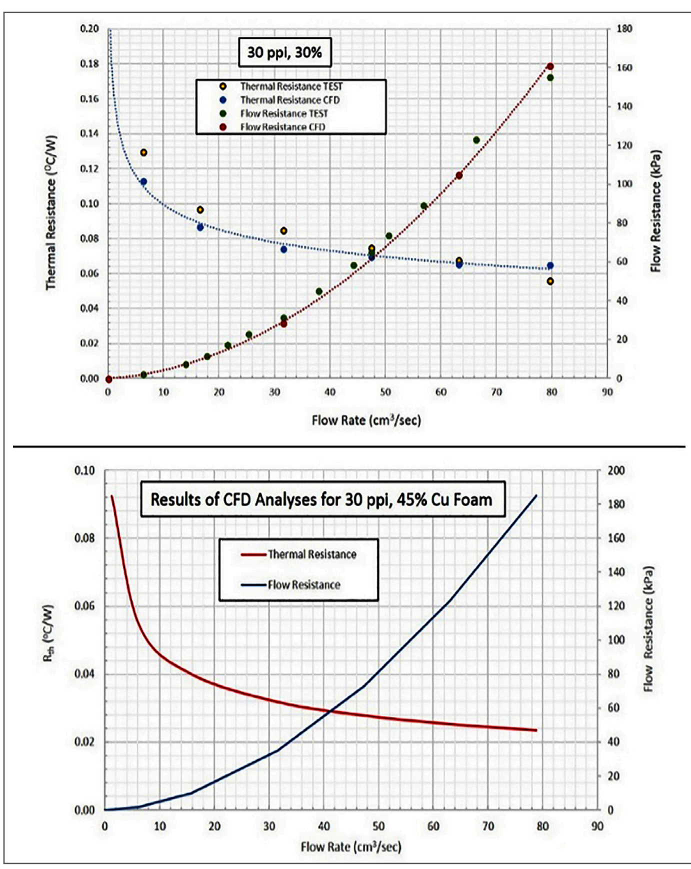

Figure 7. Flow and the thermal resistances of Cu foam based Cold plates [6].

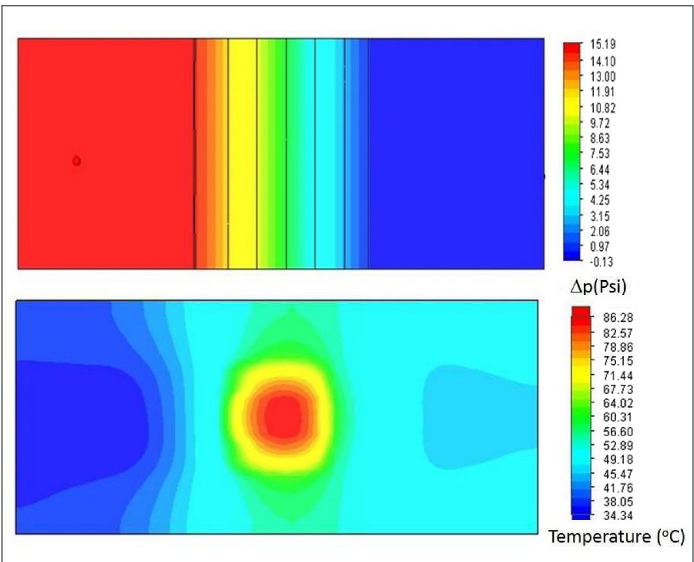

Figure 8. Distribution of differential pressure (top) and the surface temperature of the cold plate built with 30 ppi and 30% Cu RMF Foam at 1 GPM flow rate.

Effective film coefficients and surface area density of 30 ppi, 30% dense Al RMF were input to CFD using a proprietary method, to calculate the thermal and flow resistance of the test cold plate. The results of such calculations are shown in *Figures 8* for 30 ppi 30% dense RMF Al cold plate and 30 ppi 45% dense RMF foam cold plates. *Figure 8* depicts the results of CFD analyses for a cold plate built with 30 ppi, 30% dense Cu foam.

SUMMARY AND DISCUSSIONS

Studies show that RMF Based CP and HXs offer high thermal performance due to their extremely high specific surface area, local film coefficients and thermal conductivity particularly for lower volume and weight applications.

RFMs demonstrate compatibility with a wide range of liquids and gaseous coolants which makes the technology advantageously suitable for a wide range of commercial and military applications. The structural and thermal characteristics of RMF foams also offers similar advantages in passive phase change and two-phase flow applications.

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Comparison of HPC/Telecom Data Center Cooling Methods by Operating and Capital Expense

This Article Has Been Reprinted from the June, 2015, Issue

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INTRODUCTION

Current high-performance computing (HPC) and Telecom trends have shown that the number of transistors per chip has continued to grow in recent years, and data center cabinets have already surpassed 30 kW per cabinet (or 40.4 kW/m^2) [1]. It is not an unreasonable assumption to expect that, in accordance with Moore's Law [2], power could double within the next few years. However, while the capability of CPUs has steadily increased, the technology related to data center cooling systems has stagnated, and the average power per square meter in data centers has not been able to keep up with CPU advances because of cooling limitations. With cooling systems representing up to ~50% of the total electric power bill for data centers [3], growing power requirements for HPC and Telecom systems present a growing operating expense (OpEx). Brick and mortar and (especially) mobile, container-based data centers cannot be physically expanded to compensate for the limitations of conventional air cooling methods.

In the near future, in order for data centers continue increasing in power density, alternative cooling methods, namely liquid cooling, must be implemented at the data center level in place of standard air cooling. Although microprocessor-level liquid cooling has seen recent innovation, cooling at the blade, cabinet, and data-center level has emerged as a critical technical, economic, and environmental issue.

In this article, three cooling solutions summarized in *Table 1* are assessed to provide cooling to a hypothetical, near-future computing cluster. *Cooling Option 1* is an air-cooled system with large, high-efficiency, turbine-blade fans pushing air through finned, heat-pipe equipped, copper heat sinks on the blade. Rear-door air-to-liquid heat exchangers on each cabinet cool the exiting air back to room temperature so that no additional air conditioning strain is placed on facility air handlers. The water & propylene glycol (water/PG) mixture from the heat exchangers is pumped to the roof of the facility, where the heat absorbed from the CPUs is dissipated to the environment via a rooftop compressor-enabled chiller.

Cooling Option 2 uses water-based touch cooling on the CPUs via a copper cold plate loop on each board. These loops are connected to an in-rack manifold that feeds a water/PG mixture in and out of each blade. A coolant distribution unit (CDU) collects heated water via overhead manifolds from each cabinet, cools the water through an internal liquid-to-liquid brazed-plate heat exchanger, and pumps it back through the overhead manifolds to the cabinets for re-circulation. On the other side of the heat exchanger, a closed water loop runs from the CDU to a rooftop dry cooler, where the heat from the CPUs is ultimately dissipated into the atmosphere.

Cooling Option 3, which considers two approaches, removes water from the server cabinets and instead uses refrigerant (R134a) as the heat transfer fluid in the server room. This approach uses a cold plate and manifold system similar to the water-cooling approach, but may or may not have a refrigerant distribution unit in the building. *Cooling Option 3a* pumps mounted in the cabinets pump the refrigerant through a water-cooled, brazed-plate heat exchanger in a refrigerant distribution unit (RDU) to condense the refrigerant after it absorbs the heat from the CPUs. In *Cooling Option 3b*, pumps move the refrigerant straight to the roof, where a rooftop condenser dissipates the heat to the environment.

	Cooling Method		
	Option 1: Air Cooling	**Option 2: Water Cooling**	**Option 3: Refrigeration (R134a) Cooling**
CPU Cooling	Copper-fin heat sinks with embedded heat pipes	Option 2a: Series-parallel copper cold plates Option 2b: Parallel copper cold plates	Series-parallel copper cold plates
Heat Exchanger Location	Air-to-liquid on rear door of cabinet	Liquid-to-liquid in Coolant Distribution Unit	Option 3a: Liquid-to-liquid in Refrigerant Distribution Unit Option 3b: Liquid-to-air in Rooftop Condenser
Pump Location	Rooftop Chiller	Coolant Distribution Unit	Cabinet Level Pumps
Rooftop Cooling	Compressor-Enabled Chiller	Water-to-Air Dry Cooler	Option 3a: Water-to-Air Dry Cooler Option 3b: R134a Vapor-to-Air Condenser

Note: External Ambient air temperature = 25 °C

Table 1 - Overview of cooling options

The goal of this article is to provide an "apples-to-apples" comparison of these three cooling systems by suggesting a hypothetical, high-power near-future data center specification for each method to cool. When the options are compared side-by side, differences in fluid dynamics and heat transfer will translate into differences in efficiency, and comparisons between various cooling methods become more easily visible. This article is not a complete guide to installing or selecting equipment for each of these cooling systems, but it is a general overview of what power usage advantages each system offers.

HYPOTHETICAL COMPUTER CLUSTER SPECIFICATION AND THERMAL ASSESSMENT

In order to bound the comparison in a meaningful way, a set of specifications was developed from extensive discussions with the Liquid Cooling (LC) forum of LinkedIn professional network [4], an association of motivated multidisciplinary professionals from HPC, Telecom and electronics cooling industries.

After several weeks of discussion, the LC forum agreed on the following system configuration and operation conditions for analysis: the hypothetical computer cluster under consideration should produce ~1MW of IT power. The distribution of power is shown in *Figure 1*.

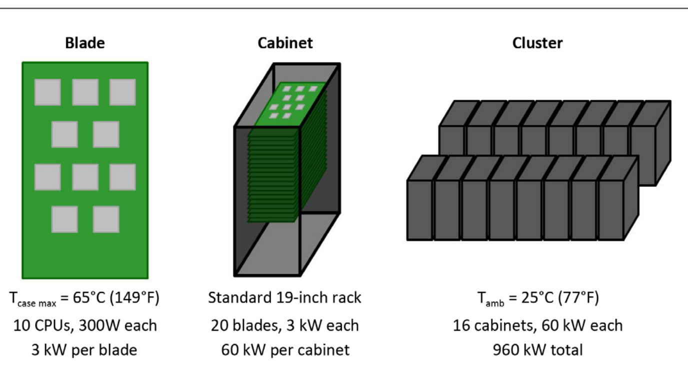


Figure 1 - Hypothetical data center module specifications

The cabinet architecture in this specification assumes each horizontal card is inserted from the front, with no cards inserted from the back of the cabinet. Alternative architectures exist, and can be cooled by a variety of means, but for this analysis, a simple, easily-relatable architecture was desired, so only front-facing, horizontal cards were considered.

A hypothetical data center could be equipped either with a dry cooler or compressor equipped chiller located on the roof of the building, at up to 60 m (≈200 feet) above computer HPC cluster level floor. To maintain the 65°C case temperature limit, the air-cooling method requires a compressor-equipped chiller, but a dry cooler is preferred for liquid cooling methods.

COOLING OPTIONS VIEWED VIA EXPLICIT COMPARISON

Option 1: Advanced Air Cooling

Cooling 60 kW in a single rack required a staggered, heatsinked 10-CPU layout, heat pipes, rear door cooling heat exchangers, and powerful, high-pressure turbine-blade fans. In addition, the air in the data center still needed to be significantly lower than room temperature (4°C) to achieve the desired case temperature. Even though air cooling may not be an economically feasible solution at this power density, and even though it would clearly not meet the NEBS GR-63 acoustic noise level standard [5], we still had to devise an air cooling option to compare to the LC options.

In this approach, each cabinet was supplied with a rear-door cold water/PG cooler with 3 "hurricane" turbine-blade fans (with operating point of ~3.7 m³/s at ~3.7 kPa pressure difference [6]). The cold water/PG solution circulates around the system to the roof chiller, where heat from the cabinets is dissipated to the ambient air. The system would be monitored with on-board temperature sensors and would either increase the fan speed or throttle CPU performance if the CPU approached the 65°C case temperature threshold. This approach is illustrated in *Figure 2*.

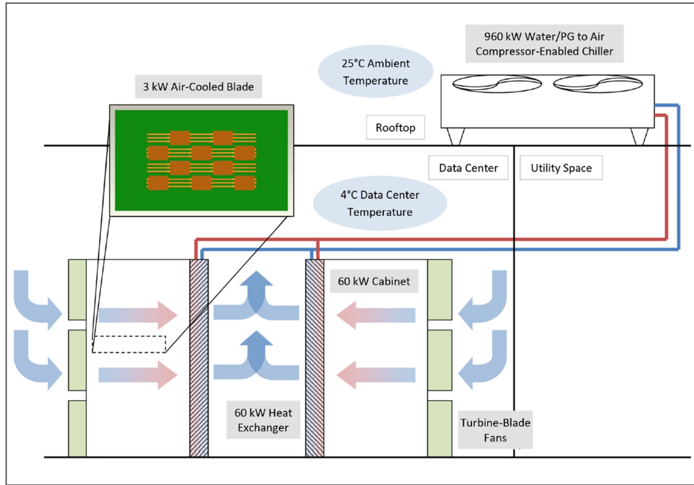


Figure 2 - Option 1: Air cooling with rear-door water/PG to air heat exchangers

Option 2: Water/PG Touch Cooling

In *Option 2a*, (*Fig.3*, top board layout) 10 CPUs per blade were ar-

ranged in two parallel groups of 5 serially connected cold plates (*Option 2a*). 20 horizontal blades were plugged into vertical supply/return manifolds, and these cabinet manifolds were supplied with water/PG from the Coolant Distribution Unit (CDU) via overhead manifolds. A pump and water/PG-to-facility water heat exchanger was needed to reduce the pressure losses at the cold plates. A separate loop brings the heat from the CDU to a rooftop dry cooler unit, where it is rejected into the ambient air. This method would include a control system in the CDU that would increase the water flow rate in the case of an increased load. Either passive or active flow regulators would also be placed at the inlet of each blade to ensure even flow distribution across the whole cabinet.

After the first pass of Water/PG system simulation it was discovered that water/PG speeds in the blade exceeded allowable ASHRAE [1] velocity limits, an additional LC option (*Option 2b*) was added - where all 10 cold plates were connected in parallel. This required additional onboard manifolds with additional piping, to allow for a compliant cold plate/CPU interface.

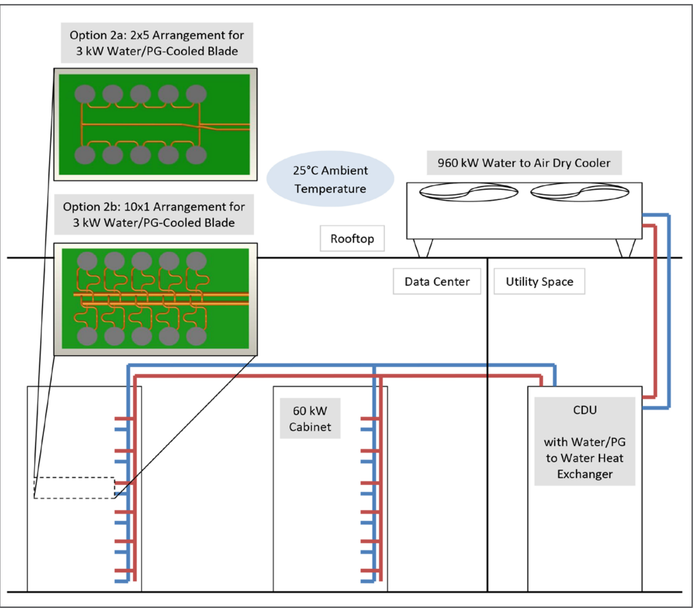


Figure 3 - Option 2: Water/PG touch cooling

Option 3: R134a Touch Cooling

The high heat of vaporization for refrigerant enables refrigerant systems to use a flow rate that is approximately 5 times less than the required flow rate for a water system with the same power. Because of this, the cold plates in *Option 3* do not need to be connected in parallel like in *Option 2b*, and can have an arrangement similar to *Option 2a*. As before, blades were connected to vertical supply/return (refrigerant) manifolds, but, because of the lower flow rate, a refrigerant pump can be fit into each cabinet to pump refrigerant through the blades and manifolds.

In *Option 3a*, the manifolds transport the refrigerant to a refrigerant distribution unit (RDU), where the heat is transferred to a closed water loop feeding into a rooftop dry cooler. In *Option 3b*, the refrigerant is sent straight to the roof, where a rooftop condenser dissipates the heat to the atmosphere. The layouts of these two options are shown schematically in *Figure 4*.

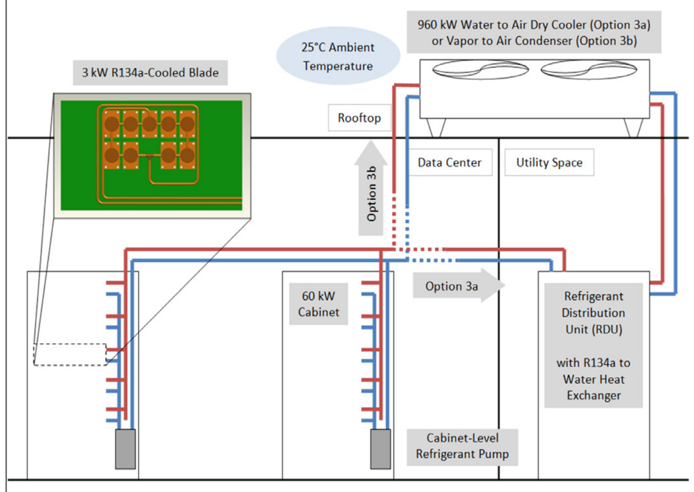


Figure 4 - Option 3: R134a touch cooling

The control system for a refrigerant cooling system includes built-in headroom to accommodate the refrigerant capacity. The system is designed so that under a full load, the refrigerant quality (the fraction of refrigerant that is vapor, by mass) does not exceed 80%, so a 20% safety factor is already built into the system at the worst-case scenario. Flow regulators at the inlet of each blade ensure even flow distribution across the entire height of the cabinet. In the event of an increase in CPU power, either the RDU would increase the water flow rate or the rooftop condenser would increase its fan speed to fully condense the refrigerant.

Capital and Operating Expenditures

With identical performance specifications (maximum case temperature and environmental ambient temperature), the differences between cooling systems can be easily compared in terms of capital and operating expenses (CapEx and OpEx, respectively). It is important to mention that the presented first-pass analysis was not intended to produce entirely optimal designs for each cooling option. For equipment selection, computational fluid dynamics (CFD) analysis [7], flow network analysis [8], a two-phase analysis software suite, and vendors' product selection software [9, 10] were used to analyze pressure drops and heat transfer across different system components.

For each cooling option, capital expenses were determined by obtaining quotes of only the main components of cooling hardware (fans, heat exchangers, pumps, cold plates, refrigerant quick disconnects, etc.) from the manufacturer, and 10% of the cost was assessed for installation, piping, etc. The cost for the electric power supply, controls, hose and pipe fittings, UPS, etc. was not included. This cost is certainly an underestimate of the total cost of installation, but it would be representative of the main cost drivers associated with each cooling system.

One important note about the CapEx estimates is that the water

and refrigerant cold plates are assumed to be of equal cost. In reality, water cold plates require higher flow rates and therefore larger tube diameters to cool the same power, but refrigerant cold plates must withstand higher pressures. The actual cost of manufacturing depends more on the manufacturing technique than it does on the fluid used, so the two cold plates were assumed to be of similar cost.

Operating expenses for all cases were calculated by determining the cost of electricity needed to pump the coolant around the loop and to run the fans. To do this, Computational Fluid Dynamics (CFD) and flow network analysis were used to calculate the pressure drop and flow rate of each fluid through the system, and then an average operational efficiency was used to determine the total power draw. This analysis assumed an electricity rate of $0.10 / kW-hr, without demand charges, and that operating hours per year were 8,760 for all methods. Since the refrigerant-based option does not require periodic flushing and replacement, in our analysis the cost of R134a was only added to CapEx. With water cooling, in order to keep electro galvanic corrosion inhibitors and microbiological growth suppressants active, water/PG Coolant mixture requires regular flushing (every 2-3 years), so the cost of water cooling additives was added to OpEx as well as the initial CapEx estimate.

Figure 6 shows that using a direct, rear-door air-cooling approach, low data center temperature, heat sinks with embedded

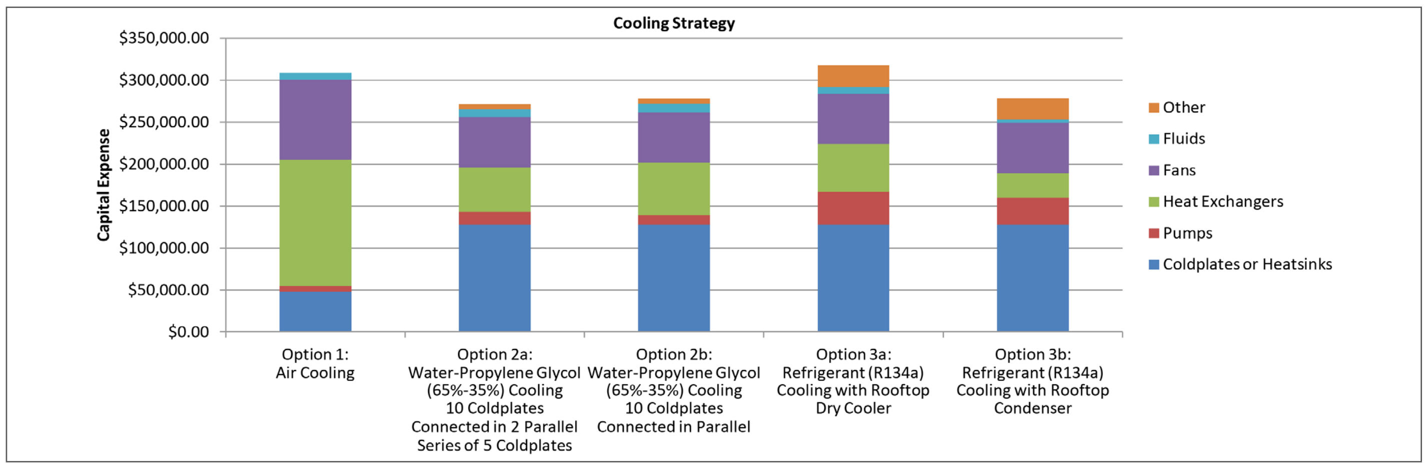


Figure 5 – Estimated capital expenses for three cooling strategies

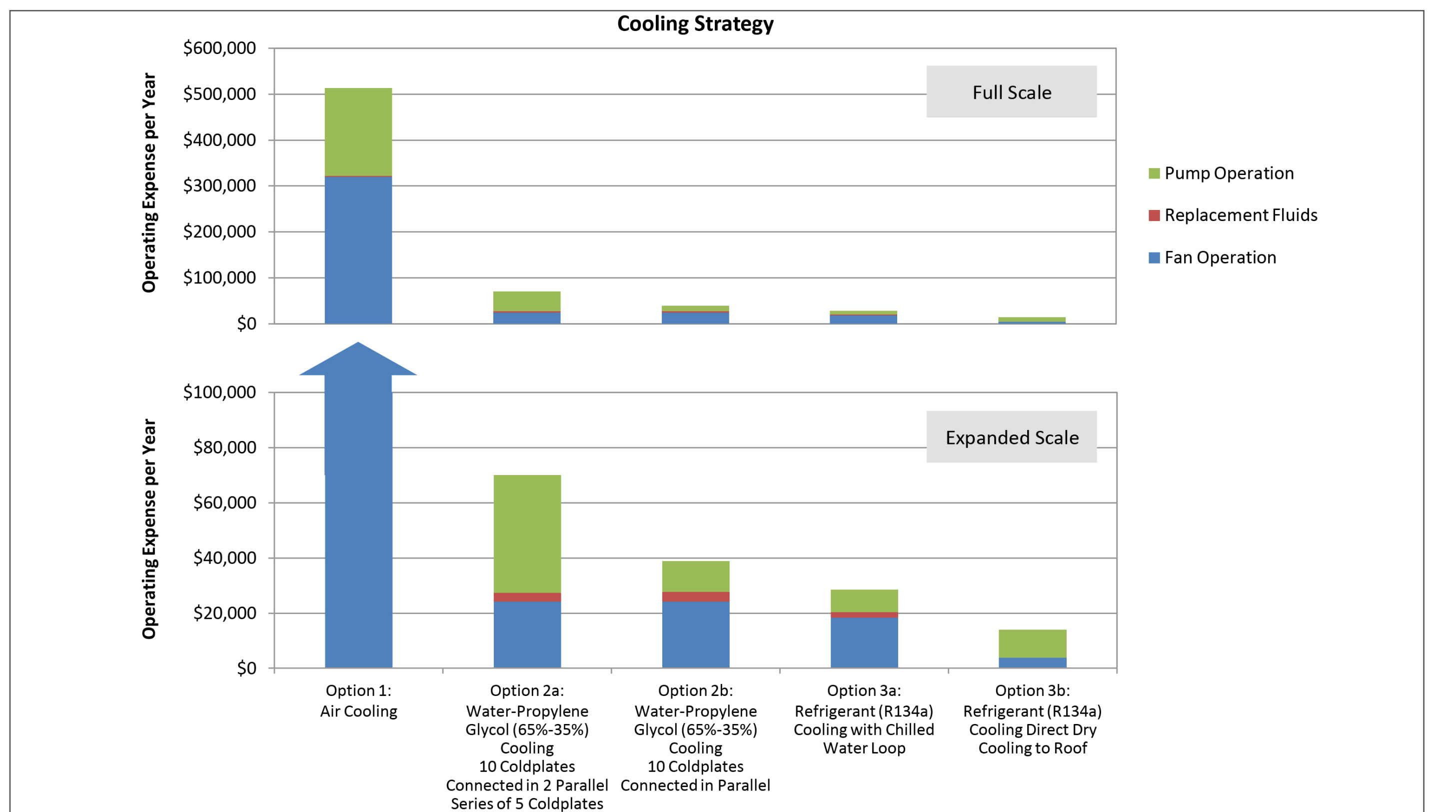


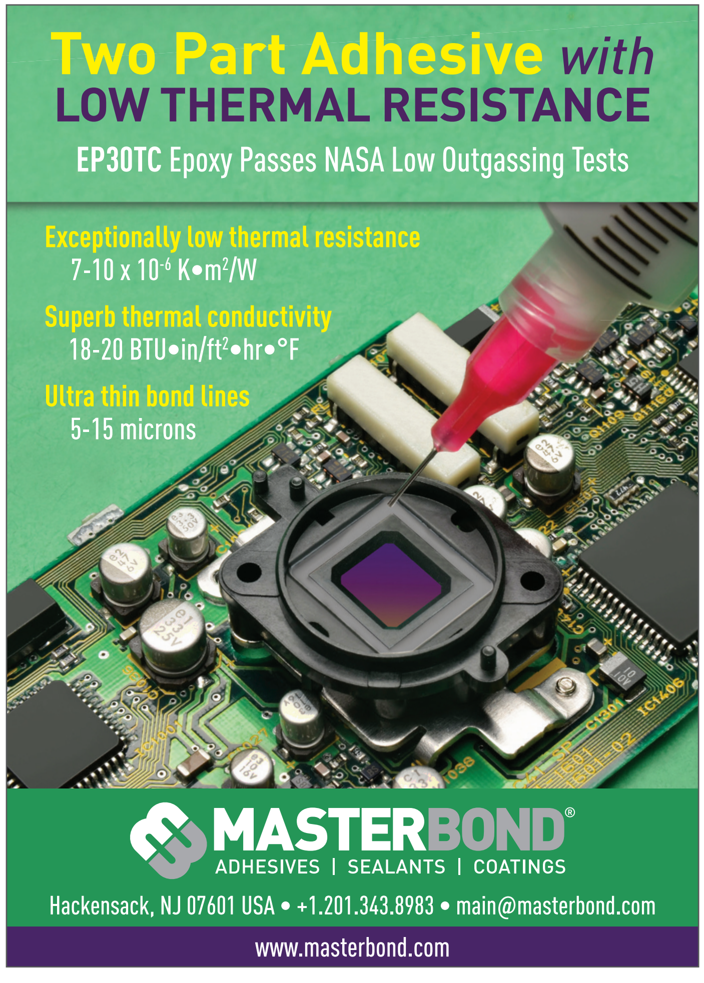
Figure 6 – Estimated operating expenses for three cooling strategies

heat pipes, and turbine-blade high-efficiency fans, air cooling will cost far more than a liquid-cooling option. Again, this best effort for air cooling option was presented only for the sake of comparison.

Figure 6 also indicates that that switching from air-cooling to any of the four liquid-cooling options will cut the operating expense (to run the cooling system, not to run the entire data center) by at least a factor of 5, In addition, the direct refrigerant *Cooling Option (3b)* shows the lowest operating cost of all the cooling options, with less than 1/30th of the cost of the air-cooled option. With this operating cost, an existing air-cooled data center (per *Option 1*) would greatly benefit from switching to a refrigerant-cooled data center (*Option 3b*), and, assuming no additional retrofit expenses, would recover the switching cost within the first year of operation.

CONCLUSION

Although the comparison in this paper is a preliminary, predictive analysis of several different cooling systems, the differences in power consumption revealed here show that a data center outfitted with liquid cooling provides a tremendous advantage over air cooling at the specified power level (60 kW per cabinet).



As HPC and Telecom equipment continues toward higher power densities, the inevitable shift to liquid cooling will force designers to choose between water and refrigerant cooling. It is the author's belief that the industry will eventually choose direct refrigerant cooling because of the advantage it has over other cooling systems in operating cost (at least 2.5 times cheaper) with similar capital cost, the minimal space requirements on the board, the absence of microbiological growth, electro galvanic corrosion, and corresponding need to periodically flush the system.

ACKNOWLEDGEMENTS

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2018 Products & Services Index

The Products & services index contains many categories to help find the products and services you need. Details of all the suppliers listed within each category can be found in the company directory, starting on page 28. To learn how to be included in this directory, e-mail editor@electronics-cooling.com

Adhesives
Rogers Corporation

Air Conditioners
TECA ThermoElectric Cooling America Corp.

Blower/Fan Accessories
Aavid, Thermal Division of Boyd Corporation

ORION Fans

Blowers
Aavid, Thermal Division of Boyd Corporation

Delta Electronics

JARO Thermal

Rosenberg USA Incorporated

Bonding
Rogers Corporation

Chillers
Aavid, Thermal Division of Boyd Corporation

Cold Plates
Aavid, Thermal Division of Boyd Corporation

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Heat Spreaders continued

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Polymer Science

T-Global

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Shiu Li Technology Co., LTD

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Laird Technologies

TECA Corporation

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