Basics of (PCB) thermal management for LED applications
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Introduction

The reader may wonder why ‘PCB’ in the title is between quotes. The original objective of the paper was to lay a firm base for people who are involved in printed circuit board (PCB) design for LED applications to understand and predict the thermal behavior of their design. However, after reading the paper the basic achievement will be the notion that such a designer cannot focus on the PCB alone, she has to take into account the rest of the world too (by way of speaking of course). Hence, the paper is equally well suited to serve all designers dealing with one or more aspects of the total LED application, be it the LED itself, the thermal interfaces, the heat sinks or the luminaire.

First of all, the paper will discuss the reasons why thermal management is important, and then treat the basics of heat transfer: conduction, convection and the concept of thermal resistance. How to perform back-of-the-envelope calculations is another topic that is covered, as well as the non-trivial concept of heat spreading, required in later stages of a design phase. Before jumping to conclusions, the paper discusses thermal interface materials and the associated wrong use of thermal impedance as their characterization.

1. Why thermal management?

The first question to be addressed is why we need thermal management in the first place. As Christian Belady (formerly HP) put it eloquently in 2001:

The ultimate goal of system thermal design is not the prediction of component temperatures, but rather the reduction of thermally associated risk to the product.

Hence, if your boss asks you: “Please take care that the junction temperature does not exceed 125 °C”, you may answer: “Why? Do we sell temperatures?” If you are not fired a good boss should reply: “Clever answer, but the reason is that the junction temperature is linked to three key issues that determine the quality of our LED-based products: Lifetime, Color point and Efficiency, and these features we do sell.” Of course, the biggest problem nowadays is to find a good boss, simply because most people become boss because they fail as an engineer.

The first step in reaching the aforementioned goal is to be able to perform back-of-the-envelope calculations to get a rough idea about the feasibility of your design from a thermal point of view. What kind of knowledge do you need to enable these calculations? Determining critical temperatures is contingent on a correct understanding of:

- Thermal conductivity $k$
- Heat transfer coefficient $h$
- The electrothermal analogue and its derivative: thermal resistance $R_h$

That’s all you need for rough calculations. However, when you need more accurate answers such as in later design stages, things become much more complicated, and the best advice is to hire a thermal expert with access to dedicated computer codes.

2. Basics of heat transfer

To understand what is required for a PCB from a thermal point of view one needs to understand the meaning of thermal conduction, convection and resistance.

Thermal conductivity (W/mK)

The notion of thermal conduction is not very old. Biot (1804) and Fourier (1822) were the first to study quantitatively the heat flow through a solid piece of material. Fourier observed that the heat flow $q$ was proportional to the temperature difference $\Delta T$ over the test piece, proportional to the cross sectional area $A$ of the bar, and inversely proportional to the length or thickness $\ell$, known as Fourier’s law:

$$q = k \frac{A}{\ell} \Delta T$$

(1)

The proportionality constant $k$ is called the thermal conductivity in W/mK. It is a material property and a measure for the ability of a material to conduct heat. The range for engineering materials is from air (0.03W/mK), via plastics (0.2 W/mK), FR4 (0.4-1 W/mK), glass (1 W/mK), aluminium (200 W/mK) to copper (400 W/mK). Typical thermal interface material (TIM) values cover the range 0.4-4 W/mK.

Heat transfer coefficient (W/m²K)

The heat generated in an electronic device is usually transported by conduction to a heat sink or an area where the heat is transferred to a fluid, which is called convection. The fluid can be a gas such as air, or a ‘real’ fluid such as water. It turns out that to first order the heat that is convected away is proportional to the area $A$ and the temperature difference between the wall and the main stream flow:
\[ q = hA\Delta T \] (2)

This equation is commonly known as ‘Newton’s cooling law’ but it should be realized that it is neither a law nor derived by Newton. The famous chap did know a lot about a lot of things but unfortunately the notion of the heat flux, let alone the heat transfer coefficient, was not part of it. The proportionality coefficient \( h \) is called the heat transfer coefficient, in \( \text{W/m}^2\text{K} \). As a rule-of-thumb, take for natural convection \( h=10 \text{ W/m}^2\text{K} \) and for fan-driven forced convection \( h=50 \text{ W/m}^2\text{K} \). It is not advised to use so-called correlations because these suggest an accuracy that is absolutely not warranted in real-life. See for more details Lasance [1].

**The electrothermal analogue and the definition of the thermal resistance**

The last term to discuss shortly is the thermal resistance which got its name thanks to the electrothermal analogue. In a DC electrical circuit, Ohm’s law describes the relations between the voltages and the currents. It states that a voltage difference over a resistor causes an electrical current, which is proportional to the voltage difference: \( \Delta V = I \cdot R \).

In steady state heat transfer, a temperature difference causes a heat flow which is proportional to the temperature difference as is seen in equations (1,2). Both equations can be written in the form \( \Delta T = q \cdot R_{\text{th}} \) with \( R_{\text{th}} \) the thermal resistance (also commonly noted as \( R \) when there is no chance for misreading it as an electrical resistance). This is analogous to Ohm’s law. In both the electrical and the thermal case we observe that a driving force exists (either voltage difference or temperature difference), which causes a flow (of current, or of heat) over a resistor. In more general terms, it appears that the differential equations describing current flow and heat flow are the same, hence the term electrothermal analogue. However, a word of caution should be issued. Differential equations alone are not sufficient, we need also initial and boundary conditions and it is here where we meet a serious problem in interpreting thermal resistances in real life.

The notion of ‘thermal resistance’ is deeply rooted in the vocabulary of thermal and electronic designers. Every textbook treats the fundamental analogy between electrical and thermal resistance, and most of the thermal data found in the Component Data Sheets are presented in the form of either thermal resistance from junction to case (\( R_{\text{th} \text{JC}} \)) or junction-to-ambient (\( R_{\text{th} \text{JA}} \)). Usually, these resistances are defined somewhere in the introduction. Of course, it is possible to define everything, and call it what you want, as long as the expressions at both sides of the equal sign have the same dimensions. There is no law that the definition should make sense from a physical point of view. And here we meet the problem. Most people are of the opinion that the definition should have a physical significance, on the grounds that an electrical resistance has certainly a physical meaning (the voltage between two points divided by the current from one point to the other), and smart professors told them that electrical and thermal differential equations are identical. Unfortunately, the conclusion (‘electrical and thermal resistance are analogous’) is wrong, while the proposition (‘electrical and thermal differential equations are analogous’) is right. Why? Mother Nature has to be blamed. Somehow, at time zero of the universe (or maybe even before that), the building stones of matter and life were arranged in such a way that what an electrical engineer calls ‘insulation’ is about 20 orders of magnitude away from what he calls ‘conduction’, while, in thermal terms, the difference between ‘insulation’ and ‘conduction’ in practice is about 3 orders of magnitude. To highlight the distinction, the thermal difference between insulation and conduction is about the difference in conduction between high-doped and low-doped Silicon in electrical terms.

A formal definition of a thermal resistance is:

*The temperature difference between two isothermal surfaces divided by the heat that flows between them is the thermal resistance of the materials enclosed between the two isothermal surfaces and the heat flux tube originating and ending on the boundaries of the two isothermal surfaces* (Rosten and Lasance [2]).

The essential point to understand is that a thermal resistance can never be based on measuring or calculating two points unless the plane is isothermal. Additionally, no heat should be lost between the two planes, see Figure 1.

![Figure 1 Two isothermal surfaces connected by a heat flux tube](image)

Now consider at a real product. *Figure 2* shows the most important features of a typical LED-based product.

![Figure 2 A typical LED-based product](image)
According to the above definition, it is formally not possible to define a thermal resistance between two points, e.g. die and case. In other words, $R_{\text{th \ die-case}}$ is only correct provided:

- The die and case surfaces are at uniform temperature
- We know the heat flux between die and case

Regarding the first bullet point: except for high-power LEDs (e.g. > 3 W) the assumption of a uniform die temperature is correct. It is the case surface that causes severe problems because the heat spreader (or alternatively the board) cannot be a priori considered to be at uniform temperature, even not for cases where the PCB is a metal-core board. This assumption should always be checked upfront. The consequence is that the measured case temperature becomes dependent on the heat transfer coefficient $h$ that describes the rate of heat transfer from the heat sink to the environment, usually including both radiation and convection.

Take the following example. For the single source case depicted at Figure 4 left a constant heat transfer coefficient $h$ at the bottom causes a non-uniform temperature profile on this face, except in the case of $h$ or the thermal conductivity $k$ being infinite, or the trivial case of the source area equalling the substrate area. The average temperature could be used, but the question in practice is: how do we get this value? Usually a single thermocouple is used at the centre, but one should realise that the thermal resistance from source to thermocouple defined in this way is dependent on the boundary conditions because these determine the temperature profile over the backplane and hence influence the temperature at the location of the thermocouple. The only metric that is in accordance with the definition is the thermal resistance from source to ambient: $R_{\text{th \ source-ambient}}$ but this value is often useless because it includes not only the heat spreading but also the air-side part that is usually not known in practice.

The second bullet point is usually also met, but should be checked in case some heat is leaking away through the optics, either by radiation directly from the source or by conduction and convection/radiation from the top surface. However, it should be stressed that in contrast to incandescent lamps the corrections with respect to radiation are of second order.

**Thermal resistances (K/W)**

In general, we can write the following equation for the thermal resistance in K/W:

$$R = \frac{\Delta T}{q}$$

The thermal resistance for conduction is (see Eq. 1):

$$R = \frac{\ell}{kA}$$

The thermal resistance for convection is (see Eq. 2):

$$R = \frac{1}{hA}$$

The unit area thermal resistance $\Theta$ (hence absolutely not thermal impedance, see Section 6) is equal to the ratio between thickness $t$ and thermal conductivity $k$ and is often used to allow for a direct comparison of the heat transfer performance of commercially available thermal interface materials (TIMs).

$$\Theta = \frac{t}{k}$$

3. How to perform a back-of-an-envelope calculation

In many cases it is good practice to start by drawing a simple thermal network using the equations explained above. Figure 3 shows the basic idea. On top the most simple network: a heat source, and a thermal resistance connecting the source temperature and the ambient temperature, for example describing the convection.

**Figure 3 top: simple thermal network, bottom: two thermal resistances in series**

At the bottom an often used network with two thermal resistances in series: one describing the heat flow by conduction, the second by convection. Often only the largest resistances deserve attention. **It is the designer’s job to find the largest resistance!**

It is clear that such a network can be expanded in both directions at will, adding successively more detail. In the end, there is no difference anymore between a very detailed network and a discretization method as is for example used in a Finite Volume Method.

For the series resistance network we can write for the temperature difference between source and ambient:

$$\Delta T = T_{\text{source}} - T_{\text{ambient}} = q \ast R_{\text{total}}$$
with: \[ R_{\text{total}} = R_1 + R_2 \]

Let’s discuss a practical example. Only the philosophy is discussed here, the calculation details are covered in a Calculator available on the web [3].

Suppose the designer knows that 5W is required to realize a certain light output. Further input she got: maximum junction temperature 120 °C, maximum ambient temperature 40 °C. The chosen LED is attached to a metal core printed circuit board (MCPCB) of area 1 cm². How to proceed? Here are the steps to follow:

- First of all, check if the 5W can be handled by the preferred type of convection (forced or natural) and heat sink (weight, volume, size, cost). To this end, sketch a network and calculate the relevant thermal resistances.
- If it turns out that 5W is too much to handle, check which thermal resistances are dominant, then focus on them.
- If the results are not trivial (such as: no problem exists even if the data are wrong by a reasonable margin) the final step should always be a detailed analysis. Recall that often we don’t talk one-dimensional heat transfer but heat spreading, which is a rather complicated issue, see Section 5.

4. The Calculator

The Calculator [3] is a spreadsheet based tool, with the following input:

- Number of LEDs
- Dimensions of LED source and PCB area
- LED thermal data from datasheets
- PCB dielectric and bulk thermal conductivity, area and thickness
- TIM thermal data between PCB and heat sink
- Area enlargement factor for heat sink
- Heat transfer coefficient to ambient air
- Maximum allowed LED and ambient temperatures
- Power dissipation

Dependent on the question, the user has a choice of options. Three examples are given below of how to estimate respectively your heat sink, convection mode and dielectric (or PCB), given a certain input.

Starting point is a given PCB area with a number of LEDs. First calculation is the area per LED. However, the user should realize that this is only valid if the dissipation of all LEDs is approximately the same. If not, you have to consult an expert. Two extreme cases are considered:

Best Case

One-dimensional heat transfer is assumed, hence no heat spreading. In other words: the LED area is equal to the PCB area.

Worst case

We assume no heat spreading in the dielectric layer direct under the LED, and ideal heat spreading in the metal part. In other words: the LED area is used for calculating the thermal resistance of the dielectric layer, the PCB area is used to calculate the thermal resistance of the metal layer.

Example 1

I need to dissipate 5W per LED for a given light output, with a prescribed LED (Luxeon Rebel in this case), a prescribed Metal Core Printed Circuit Board (MCPCB), and a prescribed thermal interface material. Per LED I have 10cm² PCB area.

**Question:** What kind of heat sink and convection mode is recommended? **Answer:**

- Even with the assumption of no heat spreading, an ideal heat sink and ideal liquid cooling (R_h = 0) the required 5W cannot be reached.
- It is clear from the data that the LED itself is the culprit.
- In this case I need an LED with a (much) smaller thermal resistance, or I should use two LEDs.

Example 2

I need to dissipate 1 W for a given light output, with the same LED-MCPCB-TIM combination.

**Question:** Can I use a standard heat sink and natural convection? **Answer:**

- The required heat sink thermal resistance is 63,8 K/W (best case) or 60,2 K/W (worst case)
- The calculated heat sink R_h is 5 K/W, due to the large area available.
- Reaching the goal is no problem at all, the area and/or heat sink can be made much smaller, or the power per LED can be raised significantly.

Example 3

Same as Example 2. Additionally, I want to use a cheap heat sink and no fan.

**Question:** What are the thermal requirements the dielectric of the MCPCB should obey? **Answer:**

- The required PCB thermal resistance is 58,9 K/W.
- Suppose the thickness of the dielectric layer (the contribution of the metal part can be neglected) of the MCPCB is 0.1 mm, then it follows that a dielectric with a thermal conductivity of 0.0017
W/mK (best case) or 0.126 W/mK (worst case) will do.

- Even cheap dielectrics have a thermal conductivity that is much higher.

Other examples, e.g. for high-power LEDs, are also discussed in the spreadsheet examples. Before jumping at conclusions, the question still needs to be addressed: do the conclusions change when we take heat spreading into account? This question will addressed in the next two sections. Based on all evidence, we may conclude the following regarding the MCPCB thermal requirements.

In the majority of the cases, especially when dealing with natural convection-driven applications, the thermal properties of the PCB are not the major bottleneck. It is often easier and cheaper to improve other elements in the thermal resistance chain.

Hence, there is often no reason to buy a MCPCB because of its better thermal performance. Of course, there may be other reasons such as CTE mismatch or breakdown voltage requirements to choose a more sophisticated PCB.

In summary, from a thermal point of view only, in many cases of practical interest it will turn out that the thermal performance of the PCB is relevant only for high heat flux cases (e.g. liquid cooling) and for top-of-the-bill LEDs.

5. Heat spreading: not a trivial issue

As argued before, designers should know upfront if heat spreading is an issue or not. Unfortunately, no simple rules exist in order to make an early decision. Unfortunately, except for the simplest of cases, the equations describing heat spreading physics do not have an explicit mathematical solution. Hence, we have to rely on clever approximations or suitable computer codes.

The following section discusses the basics of heat spreading physics. For a more in-depth discussion the reader is pointed to Lasance [4]. A distinction is made between single source and multiple sources heat spreading.

Single source

Heat spreading is essentially area enlarging: the larger the area, the more heat can be removed at the same temperature difference (subject to certain limits). Contrary to what is believed by many designers: heat spreading is not a trivial issue. Consider the simple configuration shown in Figure 4.

![Figure 4 Heat spreading from single source (left) and two sources (right)](image)

The remarkable thing is that even for this simple configuration no explicit solution is known for the description of heat spreading. Observing the exact implicit solution of the governing differential equations reveals the source of the complexity of heat spreading: it is not possible to separate the convection and conduction parts. In other words, changing the heat transfer coefficient changes also the value of the spreading resistance. Consequently, it is not possible to write the problem in terms of one conduction resistance describing the heat spreading inside the solid and one convection resistance describing the boundary condition because the two are dependent. There is one exception: the analysis becomes much more straightforward when the temperature gradients over the area that is in contact with the environment can be neglected. In other words, when a uniform temperature may be assumed. Such is often the case with relatively small heat sinks and spreaders, and when the thermal conductivity is relatively high everywhere.

A final complexity stems from the fact that decreasing the thickness of the plate does not automatically result in a decrease in temperature, caused by the fact that a smaller thickness also implies a decrease in spreading capability. Hence, for a certain combination of thickness and thermal conductivity given the boundary conditions and the dimensions a minimum in the total thermal resistance may be found.

Multiple sources

Multiple sources add another layer of complexity because the coupling between the sources is not only dependent on the dimensions and physical properties but also on the boundary conditions and, worst of all, on the dissipation of the sources themselves. Even the definition of thermal resistance is lost when more than one source is present, because the second condition for a correct definition, the fact that the same flux has to enter and leave the resistance, is violated. The essential point to understand is that when dealing with multiple sources the concept of thermal resistance becomes meaningless, except in the situation where the multiple sources/spreader assembly is subdivided into many resistances, for each of which the definition holds, being mathematically similar to a finite volume discretization.

How to address heat spreading

In growing order of complexity, we may distinguish the following four approaches to calculate heat spreading:
- 1D series resistance network with or without a geometrical correction factor
- Analytical solution-based approximate equations
- Software based on analytical solutions
- Conduction-only finite volume/element based codes

All four approaches are extensively discussed in ref. 4. For the purpose of this paper the following summary is sufficient.

For situations where one is dealing with a single source, predominantly one-sided heat transfer, and one heat spreading layer, the analytical solution-based approximate equations (easy to embed in a spreadsheet) demonstrate an order of magnitude higher accuracy over the 1D series resistance network approach.

For situations where double-sided heat transfer plays a role, or multiple sources, or multiple layers, the problem becomes intractable from an approximate analytical point of view and we have to rely on computer codes. Implicit solutions are known for multi-layer cases with multiple sources and uniform boundary conditions, even when time is a parameter. User-friendly software exists that is based on these solutions [5], with the big advantage that also people with little background in heat transfer can get insight in the physics underlying heat spreading by simply changing a few parameters. An additional advantage is that no mesh generation is required. Another recommended source of information can be found on the website of the University of Waterloo [6]. They made a number of easy to use calculators available of which one in particular is very useful for this case: the two-layer metal-core MCPCB. Also of interest: one of their papers (Culham and Yovanovich [7]) contains a couple of graphics showing clearly the errors a designer may encounter by using a simple series resistance approach. However, the user should be aware of the limitations. For more practical cases for which layers consist of more than one material or for which the boundary conditions cannot be considered uniform, more advanced conduction-only codes should be used.

Figure 3 shows some cases that can and cannot be handled by the analytical software discussed in this section.

When one is dealing with cases that resemble the case shown in Figure 5, right, one has to rely on more sophisticated codes. In principle, all Finite Volume/Finite Element/etc. codes can be used that solve the energy equation. In practice, in most cases only user-friendly codes are recommended that enable a designer to get results in an hour or so. Some popular Computational Fluids Dynamics codes used in conduction-only mode are examples of such a code. It is recommended to validate the model in an early stage by comparing the results with those obtained analytically using the software described in this section.

In summary, the author is of the opinion that using analytical solutions, including the 1D series resistance network, in one way or another has its main merits in getting insight, hence is second to none from an educational point of view. However, when accuracy is at stake in the final design stages the recommended approach for solving real-life problems is in using a 3D conduction solver.

6. The role of heat spreading in simple LED applications

Let me address now the question that was posed earlier: how important is heat spreading? Generally speaking, there is no simple answer, but for the purpose of getting insight let us take again the example of the Calculator section. We make a distinction between two cases that often occur in practice when dealing with LED applications: a single heat spreader, and a heat spreader with a thin dielectric layer on top.

6.1 Single layer

As is demonstrated in Ref.4, for this kind of heat spreading cases we may apply the following approximate equation:

\[
R_{thf-a} = \frac{1}{h_{eff} A_2} + \frac{\ln\left(\frac{A_2}{A_1}\right) - 2\gamma}{4\pi kd}
\]

where \(\gamma\) is Euler’s constant, 0.58, \(h_{eff}\) the effective \(h\) caused by the increased area of the heat sink, \(d\) the thickness of the PCB, \(A_2\) the PCB area and \(A_1\) the LED area. With some care the first term of the right-hand side could be interpreted as a convective term and the second as a conductive term. The heat spreading becomes manifest through the logarithmic term and the fact that the thickness appears in the denominator. However, a warning should be issued here: don’t use this equation before you have read the quoted paper, there are limitations to its use. The beauty of using this simple equation is that it is easily seen what the effect is of heat spreading, namely the relative magnitude of the two terms, plus the difference in the second term as compared to the simple 1D heat transfer underlying the series resistance approach discussed in Section4:

\[
R_{thf-a} = \frac{1}{h_{eff} A_2} + \frac{d}{kA_2}
\]

Let us insert some input data based on the example discussed in Section 4. The PCB area allocated to the LED is 1cm², the enhanced area by the heat sink is 20 times this

\[\text{1 Due to the negative sign this equation does not represent a simple series resistance network.}\]
area, the PCB has 1.6 mm thickness. Let us also assume that for the sake of simplicity there is no TIM, and the LED area is one tenth the PCB area. For the convective term we get: 500/h. For the spreading term we find: 60/k, for the 1D term: 16/k. What does this mean in practice? Assume natural convection (h=10 W/m²K), and k=160W/mK for the heat spreader. The convective resistance becomes 50 K/W, the spreading resistance ~0.5 K/W, and for the 1D resistance 0.1 W/mK. While it is clear that heat spreading causes the conductive part to rise significantly, compared to the convective part it is peanuts. When we switch to forced convection (h=50 W/m²K), the conclusion is the same. We should go to much higher heat transfer coefficients or much lower thermal conductivities before heat spreading becomes an issue.

6.2 Two layers

As indicated in ref. 4, the simple equation discussed above cannot be used for more than one layer. Even the far more sophisticated Song & Lee equations, adapted for two layers, cannot be used for our dielectric/metal combination because the conditions under which their use is warranted are not met. Fortunately, the aforementioned tool of the University of Waterloo does give an appropriate answer.

The Appendix provides more details about the procedure followed for our Calculator.

The most important conclusion that can be drawn from using the more sophisticated results of the two-layer heat spreading approach is that the worst-case approach discussed above is sufficiently accurate to be useful for first-order analysis. The reason is the often large source dimension/thickness of dielectric ratio. All examples given deal with a Luxeon Rebel of footprint size 3*4,5 mm.

Caveat: Be aware that the conclusions may change considerably when the size of the LED is reduced to say 2*2 mm, introducing a spreading resistance of the dielectric that cannot be neglected. In such cases it does pay when increasing the thermal conductivity of the dielectric.

7. The role of TIMs in LED thermal management

The thermal budget of many LED applications consists to first-order approximation of three parts: 1) the thermal resistance of the ensemble LED-PCB, 2) the thermal interface resistance that is defined as the sum of the thermal resistance of the interface material plus both contact resistances, and 3) the convective heat transfer to the outside world, including the heat sink. The problem in a nutshell is that in high-performance applications the interface thermal resistance can easily account for 80% of the allowed total resistance.

Figure 6 shows a TIM between two materials, and it is clear that the effective thermal resistance consists of the TIM plus two contact resistances. The effective material thickness is called the bond line thickness (BLT). To enable an optimal choice the thermal performance of TIMs should be known with certain accuracy. The main problem is the often unknown contact resistance. Measurements are in fact the only choice because no theory exists that predicts the value with the required accuracy despite serious progress in the science of contact resistance.

Lasance [8] and Lasance et al. [9] discuss the main problem with TIM characterization: the often significant difference between standard tests performed by TIM manufacturers and real-life tests. It is instructive to summarize the parameters that are application-specific and may influence the thermal behaviour of the TIM in the final product.

- In factory assembly applications, the inability to measure the thickness and actual TIM material quantity applied with materials such as thermal greases,
- Flatness and surface conditions of both heat sink and component,
- Pressure applied. The current ASTM D5470-01 standard prescribes a pressure of 3 MPa, far above what is used in practice (0.1 MPa) (in the next revision the standard will permit for lower pressure testing),
- Clip clamping force or screw torque,
- Clip installation,
- Time-dependent phenomena, e.g. reduction in clamping force, warpage, ageing of TIM, pump-out of siloxanes or other fluids or carrier constituents,
- Non-uniform surface heating,
- Thermocouple placement. The best but also most difficult method is to measure the case temperature of the package at the hottest spot,
- Presence of manufacturing machining oils, solvents, washing agents, plastic injection molding release agents, or other contaminants present on volume-manufactured components,
- Problems increase with higher thermal conductivity (the future direction), because the influence of the contact resistances becomes larger.
The conclusion is that it is very difficult or even impossible to reproduce operational contact resistances in a standardized test method, for the simple reason that the vendor cannot possibly know what the application will be. However, the conclusion should not be that henceforth the standard test should not be performed. After all, the vendors are responsible for the characterization of their materials, which should include information of some reproducible contact resistance. It is the responsibility of the user to address the application-specific contact resistance issue.

**Reliable** vendor data should be interpreted as the minimum value a customer can possibly acquire, given a certain pressure. It is the responsibility of the designer to estimate the operational value and judge the relative magnitude of the TIM and contact resistance contributions.

In a paper from 2005, Maguire et al. [10] did a series of tests with a high-power amplifier on an extruded heat sink and demonstrated clearly the huge differences between vendor data and field data. Greases, gap pads, PCMs and some homemade compounds were compared. The vendor data and the field tests showed differences of a factor of two. Even the ranking was different. In all cases, the vendor data underestimated the real-life interface resistance.

**Thermal impedance??**

It is important that all people involved use the same terminology to define the performance of TIMs. The problem is that part of the people (mostly vendors) uses the word 'thermal impedance' as shorthand for ‘unit area thermal resistance’. This violates the electrothermal analogy commonly in use because of two reasons. First, in the electrical world 'electrical resistance' and 'electrical impedance' have the same unit, namely Ohm. Consequently, 'thermal impedance' should have the dimension K/W, not K m²/W. Second, 'electrical impedance' is a time-dependent quantity. In limiting cases, for frequency zero or large enough times approaching steady state, the impedance becomes equal to the resistance. Sticking to the current definition of 'thermal impedance' will cause a lot of confusion in the future, because the use of dynamic test methods is the obvious choice for application-specific tests, one output of which is thermal impedance. When quoting the performance of a TIM per area, we propose to use ‘R-value’ (universally accepted in the building field), 'unit area thermal resistance' or simply 'unit thermal resistance'.

**Conclusions**

Regarding thermal management of LED applications it is demonstrated that the designer needs at least a first guess about the thermal behavior of the whole system, even if she is only responsible for a single part, such as the PCB or the heat sink. After discussing the essentials of heat transfer, a spreadsheet-based Calculator is demonstrated to facilitate the assessment of the dominant thermal resistances in a series network comprising the thermal path from junction to ambient. It is shown that in many practical cases the thermal properties of an MCPCB don’t play a significant role, but that it makes sense to use the heat spreading part of the Calculator available through ref. [3] to check this rule-of-thumb. When accuracy of temperature prediction is required in later design stages the designer has to rely on more sophisticated tools.

Finally, the important role of thermal interface materials is treated, including the reasons for the often doubtful thermal data published by the vendors, and concluded that their use of the word ‘thermal impedance’ should be forbidden by law.

**References**


**Appendix**

As pointed out in ref. [4] the approach for extending the one-layer equations to two layers is in calculating an effective heat transfer coefficient and area for the second layer to be used as a boundary condition for the first layer. The problem with both the S&L equations and the UoW tool is that they result in the right answer for the total thermal resistance, but not for the individual contributions of the spreading thermal resistance and the convective thermal resistance. The reason is that the convective resistance is based upon the total area. This is the right approach for a
metal spreader, but not for a thin dielectric layer, as can easily be seen when considering a small heat source on a thin layer with low thermal conductivity. This causes a hot spot of about the same size as the heat source on the other side, which is the area available for heat transfer. Obviously this thermal resistance is much higher than when taking the whole area.

For our calculator we need this split, because we want to see all elements of the chain in order to make the right design decisions. The method chosen is the following: for the dielectric we use a spreading resistance based on some spreading angle rule, from which also the source area follows that is subsequently used to calculate the heat spreading in the metal using the S&L equations.

**Validating the Calculator**

We found that the best angle rule that matched the UoW results was about half the famous 45° spreading angle rule (see e.g. Guenin [11]. The UoW tool provides only the average results, not the maximum ones. This is a reasonable assumption for our case because the LED is represented by a flat heat source, but in reality the LED itself acts as a kind of heat spreader, hence averaging out the gradients. For a whole range of variables our Calculator matched the UoW results to within 5%. The author found it remarkable that the other tool that is based on the same set of equations (ref. 5) and which can handle also maximum temperatures, multiple layers, arbitrary sources and transients, generated average results that differed from the UoW by more than 10%. No explanation has been found so far.