Chapter

Heat Exchangers for Electronic Equipment Cooling

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Abstract

Recent developments in the electronic equipment market have been very demanding on two important design parameters: the size of the equipment and the efficiency of the cooling system. Indeed, the race for more applications handling in reduced sizes in the case of smartphones requires the use of important amounts of energy in tiny volumes. Similar constraints are encountered in the design of the new generation of vehicles (electric cars, hybrid vehicles, high-speed trains, airplanes), which impose the use of highly integrated electronic structures, resulting in significant power densifications (up to several hundred Watts/cm^2). In some CPU boards, the power generated per unit chip area is in the order of 500 kW/m^2. Cooling of such boards requires low volume and lightweight heat exchangers to transfer tremendous amounts of heat. The same situation is encountered for most newly developed demotics' equipment. This chapter reviews available state-of-the-art technologies for electronic equipment cooling, including *passive* and *active techniques*, as well as *one and two-phase heat exchange*. Directions for the design of the different heat exchangers will also be given.

Keywords: passive cooling, active cooling, spray cooling, refrigerated cooling, heat pipes, cold plates, vapor chambers, microchannel heat sinks, thermal resistance, carrying capacity, thermal program, effective thermal conductivity

1. Introduction

Operation of active electronic components (transistors, integrated circuits, microprocessors, CPUs, GPUs, etc.) generates significant amounts of heat, around 500 kW/m^2, quite comparable to the flux densities encountered at the nose of a space shuttle entering the atmosphere.

If this heat is not extracted, it leads to important increases in the temperature of these components. Such increases ineluctably induce degradation of the bonding wires, delamination of solders, and/or the appearance of leakage currents. Moreover, temperature gradients between electronic chips and their soles generate cyclical thermomechanical stresses throughout the lifetime of the chips, which results in thermal fatigue, leading ultimately to component failure.

Thus, one of the problems with the operation of active electronic components is energy dissipation; How can an electronic system get rid of the heat its operation generates? Usually, for each electronic component, manufacturers specify a maximum operating temperature (the maximum junction operating temperature, T_{op}).

If they are not properly cooled when operating, electronic components will certainly reach T_{op} and probably go over it. They then can completely lose their properties, which could lead to an alteration of the operation of the circuit boards or the systems they compose. This alteration can manifest through a complete breakdown of the systems or through a shortening of the mean time between failures (MTBF), leading to a premature aging.

Consequently, design techniques of electronic systems always consider a built-in cooling system, often called a *heatsink* to allow the component to get rid of the generated energy by dissipating it toward its surroundings. Heat sinks are actually special *heat exchangers*; they ensure the evacuation of the heat generated by the operation of electronic components.

This chapter reviews state-of-the-art technologies available for electronic equipment cooling. These technologies include *passive* and *active*, as well as *one and two-phase heat exchange*. Directions for the design of the different heat exchangers are also be given.

2. Passive heatsink technologies

Passive heat sinks are heat exchangers in the form of finned radiators, generally made of aluminum. They are called *passive* because they do not have any moving mechanical component (fan) designed to force airflow. Air moves through them only due to density difference: Hot air goes upward as it is replaced by cooler air. It is for this reason that passive sinks transfer heat to air essentially by natural convection[1]. This type of heat sinks is, by far, preferred for cooling electronic systems [2]; they are cost-effective, simple to find and assemble, and they generate no power consumption or noise.

The most widely used passive heat exchangers are *finned natural convection heat sinks*.

2.1 Finned heat exchangers

In their simplest version, these heat sinks are constituted of finned surfaces. **Figure 1** shows examples of *finned heat exchangers* used as heat sinks on electronic components. They are usually constituted of materials having a high thermal conductivity, such as aluminum or copper. But other new materials, like ceramics, can also be found.

The sink is usually mounted on the electronic component, which generates heat, thus increasing its heat transfer area[2] with ambient air. **Figure 2** shows a transistor mounted on a finned heatsink.

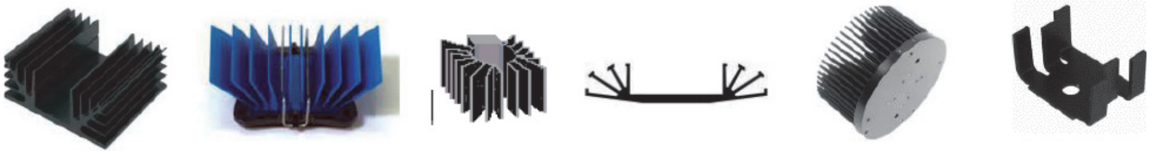

Figure 1.
Examples of finned heat sinks. Sources [3, 4].

[1] For a better understanding of natural convection see Ref. [1].

[2] The heat transfer area is the surface thru which energy is transferred from the electronic component to the environment.

Figure 2.
A heatsink fixed on a transistor. Source [5].

It can be seen from this figure that this type of mounting permits to increase the heat transfer area; the heat generated by the transistor is transferred to ambient, not only through the external surface of the transistor but also through the surfaces of all the fins.

In such situations, the heat transfer process depends on several parameters such as the temperature difference between air and the electronic component, the total area of the fins and their position (vertical/horizontal), the fins spacing, etc.

2.2 Design fundamentals

2.2.1 Thermal resistance

Despite this complexity of the heat transfer process, design techniques usually use a simple model to represent the flow of energy through a heatsink. This simple model is based on a similarity between the way that heat moves from one medium to another and the way that electric current flows from one potential to another. We know that if a point at a potential V_0 (**Figure 3a**) is separated by an electrical resistance R from a second point (at V_1), and then an electric current i flows between these two points, such that:

$$i = \frac{V_0 - V_1}{R} \tag{1}$$

Similarly, it can be shown that the power, Q, of heat dissipation between a point at temperature T_c and a point at temperature T_a (**Figure 3b**) is given by:

$$Q = \frac{T_c - T_a}{R_{th}} \tag{2}$$

Where R_{th} is called the *thermal resistance* between these points [6].

R_{th} is expressed differently for conduction and convection as follows:

- For a conduction heat transfer:

$$R_{th}^{cond} = \frac{L}{kA} \tag{3}$$

Figure 3.
Thermal resistance model for heat flow. (a) Electric current i *flows between* V_o *and* V_1*. (b) Heat flux* Q *flows between* T_c *and* T_a*.*

Where L is the material thickness, k is the thermal conductivity, and A is the heat transfer area.

- For a convection heat transfer:

$$R_{th}^{conv} = \frac{1}{hA} \tag{4}$$

Where h is the convective heat transfer coefficient and A is the heat transfer area.

2.2.2 Electronic component without a heatsink

Consider an electronic component with junction temperature T_j, in an ambient environment (air for example) at T_a.

As shown in **Figure 4**, the heat generated by the component is transferred by conduction from the junction to the external surface of the case. Then, heat is conveyed by convection and radiation to the ambient environment.

For normal operating temperatures, conduction and convection are the prevailing modes. The total heat transfer resistance is therefore given by the following:

$$R_{JA} = R_{JC} + R_{CA} \tag{5}$$

Where:

R_{JA} is the junction to ambient resistance.

R_{JC} is the conduction resistance.

R_{CA} is a convection resistance.

Note that R_{JC} is generally quite low, but R_{CA} is usually high enough to limit heat transfer from the component to environment.

2.2.3 Electronic component with a heatsink

Figure 5 shows the electronic component of **Figure 4** to which a finned heatsink was added in order to take advantage of its large contact area with ambient, thus permitting a better spread of the heat generated by the component.

In this case, as well, heat is transferred from the component to the ambient essentially by conduction and convection. Energy transfers are therefore represented by a series of thermal resistances:

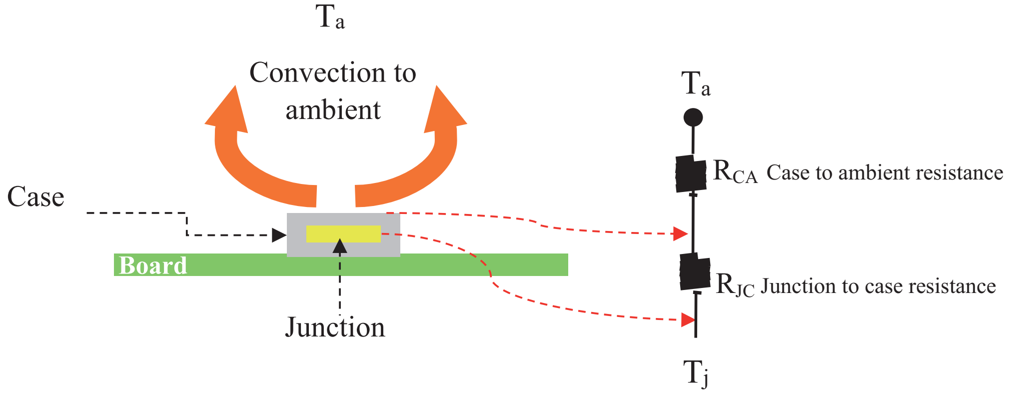

Figure 4.
Heat resistance of a sink.

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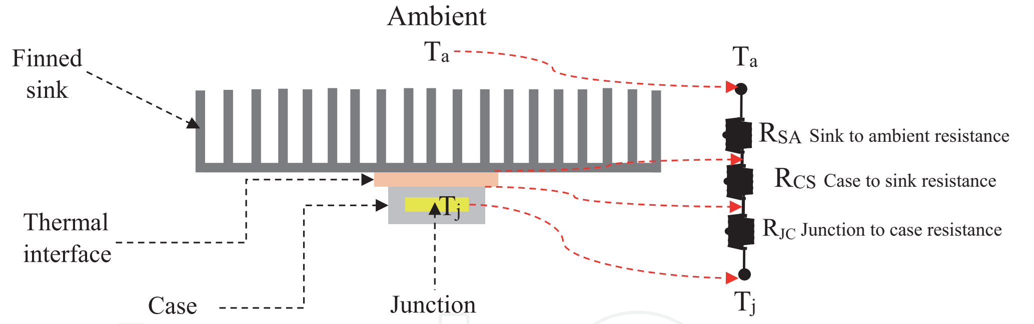


Figure 5.
Total resistance of a heatsink.

R_{JC}, *the junction to case thermal resistance*; a conductive resistance which depends on the thickness, e_{JC} (between the junction and the thermal interface), on the area S_{JC}, and on the thermal conductivity k_{JC}. R_{JC} is given by the following:

$$R_{JC} = \frac{e_{JC}}{k_{JC} S_{JC}} \quad (6)$$

R_{CS}, the case to heatsink thermal resistance; a conductive resistance that takes into consideration the thickness and conductivity of the case and interface material. R_{CS} is given by the following:

$$R_{CS} = \frac{e_{CS}}{k_{CS} S_{CS}} \quad (7)$$

R_{SA}, *convection resistance* between the fins and air. R_{SA} is a convective resistance given by the following:

$$R_{SA} = \frac{1}{h S_F} \quad (8)$$

Where S_F is the fins surface area in contact with ambiance and h is the convection heat transfer coefficient between the fins and the ambient environment (see Appendix 1).

The sum $R_{JC} + R_{CS} + R_{SA}$ represents the *heat exchanger thermal resistance* between the junction and ambient. *The thermal resistance of the heat exchanger* is then given by the following:

$$R_{th} = R_{JC} + R_{CS} + R_{SA} = \frac{e_{JC}}{k_{JC} S_{JC}} + \frac{e_{CS}}{k_{CS} S_{CS}} + \frac{1}{h S_F} \quad (9)$$

Thus, the *heat exchanger thermal resistance* is a parameter, which depends on the materials constituting the heatsink, the casing, and the thermal interface material. It also depends on the surface area in contact with air, the configuration of the fins, their number, the position of the heatsink (horizontal/vertical), air temperature, etc. Its determination is somewhat complicated [6] and often necessitates the running of experiments.

Hopefully, when integrated systems are offered, values of the thermal resistance, R_{th}, are given by manufacturers' data. In the case where the heat exchanger is mounted from separate pieces, the heat exchanger thermal resistance can be computed from Eq. (9). Electronic component suppliers' data sheets give the parameters necessary for the computation of R_{CS}. Similarly, characteristics of gaskets

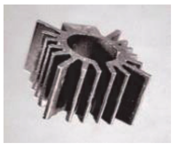
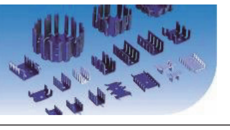
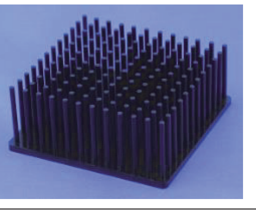
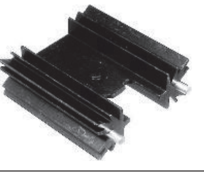
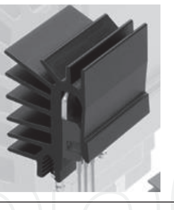
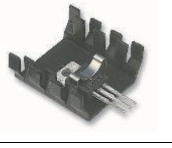
N°	Source	Heatsink	Thermal resistance *R* (°C/W)
1	[7]		R_{th} = 2.4°C/W
2	[8]		R_{th} = 17°C/W
3	[9]		R_{th} = 28°C/W
4	[10]		R_{th} = 55°C/W
5	[11]		R_{th}: 7–80°C/W
6	[12]		R_{th}: 2–56°C/W
7	[13]		R_{th} = 11°C/W
8	[14]		R_{th} = 4.75°C/W
9	[15]		R_{th} = 63°C/W
10	[16]		R_{th} = 14°C/W

Table 1.
Values of R_{th} *for different heat sinks.*

(eventually), gap fillers, and other interface materials will permit the calculation of R_{CS}. Finally, R_{SA} can either be taken from heatsink data sheets (**Table 1** shows values of R_{SA} for different heatsink makes), or calculated from Eq. (8) and Appendix 1.

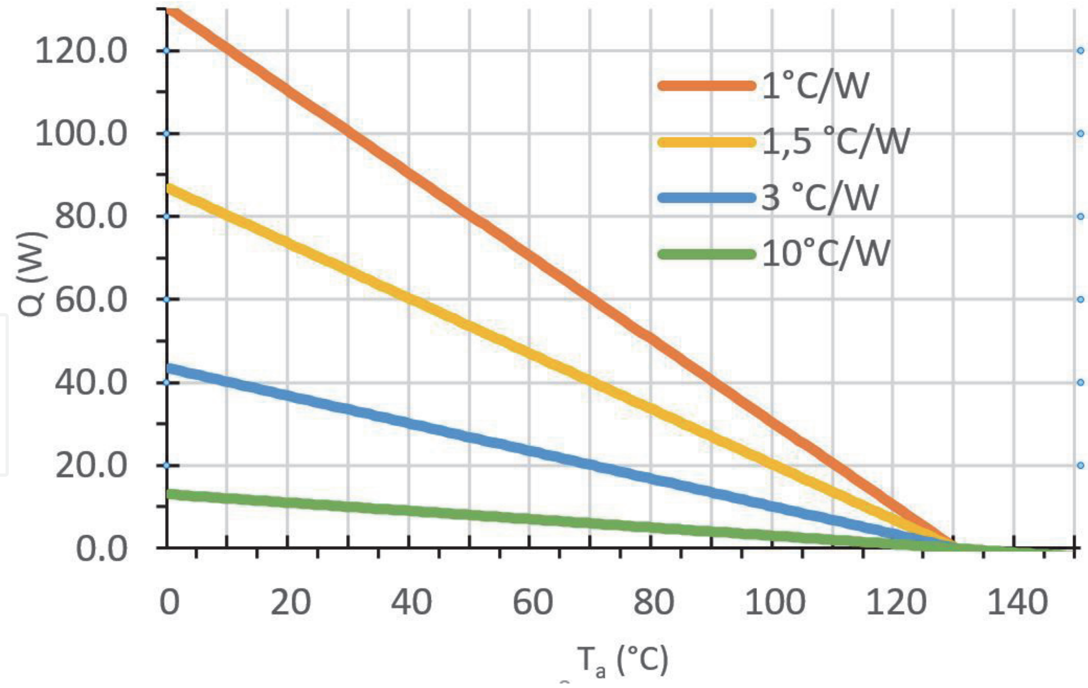


Figure 6.
Dissipation power as a function of ambient air temperature.

Remarks:

i. Sometimes, some manufacturers give two values of the thermal resistance of the heatsink. This can be the case in two situations:

- If the heatsink is such that it can be mounted vertically or horizontally, the lower value of thermal resistance corresponds to the *vertical mounting* of the fins, which does not get in the way of air movement from the low to the high parts of the fins. Heat transfer is therefore more efficient. In contrast by opposing air movement, horizontal assembly of the fins leads to a 20% of efficiency.

- If the gasket and the filling material resistances are not included, the second value corresponds to the resistance of these mounting accessories. In this case, one should add the two values to obtain the thermal resistance, $R_{CS} + R_{SA}$.

ii. In this chapter, R_{th}, refers to the total resistance of the *heat exchanger*, as shown in Eq. (9).

iii. It should be noticed that the smaller the *heat exchanger thermal resistance* is, the higher is the *dissipation power* Q, and the better is the heat exchanger and the cooling of the electronic equipment. **Figure 6** shows the evolution of the dissipation power as a function of the surroundings temperature for a component with T_{op} = 130°C, using different heat exchangers: R_{th} = 1°C/W to R_{th} = 10°C/W

3. Passive heat exchanger design

The simplest design technique uses the thermal resistance model of the heat exchanger. It supposes that the coolant (air for example) temperature, T_a, is constant in contact with the heatsink. Using this model, the power of heat dissipated by a heatsink is written as follows:

$$Q = \frac{T_J - T_a}{R_{th}} \tag{10}$$

Where:
Q is the heat flux from the component to ambient air (Watts),
T_J and T_a are, respectively, junction and air temperatures (°C),
and R_{th} is the thermal resistance of the heat exchanger (°C/W).
Three design problems can generally be encountered:

- *A sizing problem:* Given an electronic component or system with a maximum power generation P, how can one choose, among commercially available products, the heatsink that would ensure dissipation of the heat generated during operation?

- *A verification problem:* You have an electronic component/system with a heatsink mounted on. Is this heatsink correctly sized; that is, will it fit your electronic component or system?

- *Positioning circuit boards on a rack for optimal heat exchange:* How can a number of circuit boards are placed in a rack so that cooling is optimal?

3.1 Heat exchanger sizing

In this case, it is desired to find a commercially available heatsink, capable of dispersing the heat generated by the operation, in surroundings of temperature T_a, of an electronic component with a maximum power generation Q_{max} and a maximum junction operating temperature, T_{op}.

The sizing procedure is carried out in two steps.

STEP 1: From component, thermal interface, and heatsink datasheets select those which satisfy the space constraint. Get the values of Q_{max}, T_{op}, R_{JC}, R_{CS}, and R_{SA}.

STEP 2: Write Eq. (10) for Q_{max} and determine the thermal resistance of the heatsink which will insure operating temperature not exceed T_{op}.

Written for Q_{max} and T_{op} Eq. (10) gives the following:

$$Q_{max} = \frac{T_{op} - T_a}{R_{th}} \tag{11}$$

Rearranging, we obtain:

$$R_{th} = \frac{T_{op} - T_a}{Q_{max}} \tag{12}$$

Illustration 1

An electronic chip is required to operate at a surrounding temperature of 45°C. This chip is mounted with a casing having a thermal resistance R_{JC}, and with thermal interface materials having a global resistance R_{CS}. The data sheet of the chip shows that it can generate up to Q_{max} at T_{op}.

*Supposing there are no space or geometric constraints, what are the heat sinks that can be used among those of **Table 1**.*

Data:
$R_{JC} = 0.3°C/W$ $R_{CS} = 0.4°C/W$.
Q_{max} =*10 W* T_{op} = *85°C.*
Solution
Using Eq. (12), the thermal resistance at maximum power is: $R_{th} = \frac{85-45}{10} = 4°C/W$
The adequate heatsink resistance is obtained using Eq. (9) as follows:

$$R_{SA} = R_{th} - (R_{JC} + R_{CS})$$
$$R_{JC} + R_{CS} = 0.7\ {}^{\circ C}\!/_{W} \Longrightarrow R_{SA} = 3.3\ {}^{\circ C}\!/_{W}$$

This means that only heat sinks with thermal resistances lower or equal to 3.3°C/W will be able to give heat exchangers with $R_{th} \le 4°C/W$, *thus ensuring safe operation of this chip.*

This is the case for references 1 and 6 only.

3.2 Heat exchanger verification

In this case, it is desired to verify if a given heatsink will permit the safe operation of an electronic component at a known surroundings temperature.

The verification procedure is carried out in four steps.

STEP 1: From data sheets (component, thermal interface, and heatsink) get the values of Q_{max}, T_{op}, R_{JC}, R_{CS}, and R_{SA}.

STEP 2: Calculate R_{th} from Eq. (12): R^*_{th}.

STEP 3: Calculate R^*_{SA} from Eq. (13) as follows:

$$R^*_{SA} = R^*_{th} - (R_{JC} + R_{CS}) \tag{13}$$

STEP 4: If $R_{SA} \le R^*_{SA}$ then the heatsink will do the job.

Illustration 2

You bought an electronic component with a heatsink mounted on, thus constituting a heat exchanger ready to use. The heatsink is the one shown under N° 10 in ***Table 1****.*

The data sheet of the electronic component gives the following information:
Q_{max} = *10 W and* T_{op} = *115°C.*

Will this system operate safely at a surrounding temperature of 27°C?

Solution

STEP 1: Q_{max} = *10 W,* T_{op} = *115°C and*
N° 10 in ***Table 1*** *gives:* R_{th} = *14°C/W.*
STEP 2: Eq. (11) gives: $R^*_{th} = \frac{115-27}{10} = 8, 8°\frac{C}{W}$.
STEP 3: $R_{th} > R^*_{th}$*: the mounted heatsink will not do the job.*

3.3 PCBs optimal assembly

A typical passive heat exchange problem is encountered when one needs to assemble a set of circuit boards on a rack.

Figure 7 shows a number of boards of the same size, H, placed on a rack of length L and width w. The spacing between the different circuit boards is assumed to be constant, δ, whereas each board releases a constant flux density, q (W/m^2). The cooling of the circuit boards is ensured by natural convection.

In such a configuration, the temperature of the boards is not uniform, rather it increases with height, reaching its maximum value at H. The spacing δ needs to be such that heat evacuation is optimum.

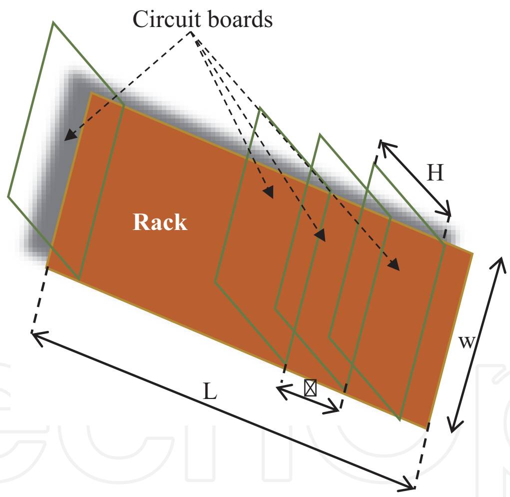


Figure 7.
Assembling a set of circuit boards on a rack.

3.3.1 Optimum spacing of PCBs

A frequent question is; *how should the boards be placed on the rack for optimum heat evacuation?* In other words, *what is the optimum spacing,* δ^**?*

The optimal spacing is given by [17]:

$$\delta^* = 2,12\frac{w}{Ra_q^{0,2}} \tag{14}$$

Ra_q being the Rayleigh number for, q, given by:

$$Ra_q = 2,12\frac{w^4 C_p \rho^2 g \beta q}{\mu k^2} \tag{15}$$

Where

- C_p, ρ, β, μ and k are respectively the specific heat, the density, the expansion factor, the viscosity, and the thermal conductivity of air[3].
- g gravity constant; g = 9.81 ms^{-2}

3.3.2 Optimal number of boards

The optimum number of boards of thickness e, to be assembled on a rack of length L, is then given by:

$$n^* = \frac{L}{\delta^* + e} \tag{16}$$

Other issues of importance in rack design such as the determination of heat transfer coefficients between electronic boards and air, and the calculation of the heat flux evacuated by natural convection are discussed in Ref. [1].

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[3] See Data Bank in Ref. [18].

4. Active heatsink technologies

During the last decade, unprecedented technological developments imposed tremendous flows of information (data or multimedia), implying an *ever-growing need for fast data transfers* (Bluetooth, Wi-Fi) and for *larger computing and storage capacities*. In addition, the advent of 5G and the *Internet of Things* means that the number of connected devices is constantly increasing [19]. Moreover, *onboard electronics* and *mobile telephony* created new ergonomics and space (volume) constraints. From a systems engineering perspective, this implies that *larger power circuits have to be integrated in smaller volumes*.

Under such conditions, the amounts of energy generated by electronic circuits are so great that *no passive heatsink of a reasonable size will be able to do the cooling job*. This is because passive sinks dissipate heat mainly through natural convection, but natural convection is no longer sufficient to extract the heat generated by the operation of these power components. Thus, the use of active cooling systems, with or without refrigeration or phase change is necessary [20, 21] for the safe operation of power electronics.

Active cooling systems are heat exchangers where the flow of the heat transfer medium (air or liquid) is forced by a fan or a pump. Several technologies exist. They can be sorted into two categories:

i. single-phase heat sinks, and

ii. two-phase heat sinks.

4.1 Single phase

Single-phase heat sinks are heat exchangers in which the cooling fluid (liquid or gas) *does not undergo any phase change*, meaning that if the cooling fluid is a liquid, then it remains liquid throughout the cooling process.

Single-phase heat exchangers used for electronic systems cooling include the following:

- Forced convection heat sinks
- Cold plate heat sinks
- Microchannel coolers
- Refrigerated coolers

By far, forced convection heat sinks are the simplest.

4.1.1 Forced convection heat sinks

Forced convection heat sinks are typically heat exchangers formed of finned surfaces similar to those presented in **Table 1**, where cooling air is forced through the fins using *fans*.

Figure 8 shows such a heat exchanger fixed on a CPU board. The fan is mounted on top of the heatsink to draw air through the fin surfaces.

Depending on the type of fins configuration, the fan can also be mounted in the middle of the finned heatsink as shown in **Figure 9**.

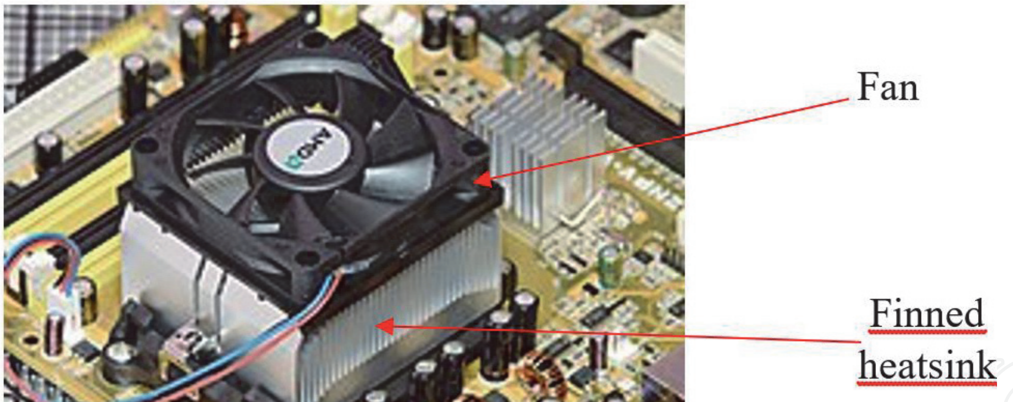

Figure 8.
A fan on a heatsink mounted on a CPU board. Source [22].

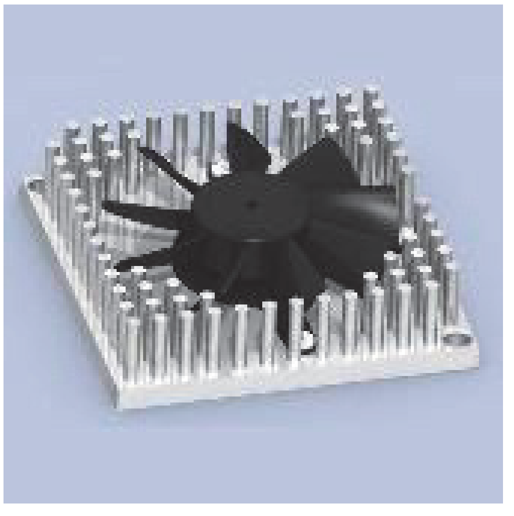

Figure 9.
Fan in the middle of a finned heatsink. Source [23].

Figure 10.
Miniaturized forced convection heatsink module. Source [24].

Very often, the assembly fins-fan is miniaturized in a cooling module similar to the one presented in **Figure 10**.

Forced convection heat exchangers become necessary when Q_{max} exceeds 70 W/cm^2 [25]: Using a fan to force air through the finned heatsink increases significantly heat transfer to air; *convective heat transfer coefficients* can reach values as high as 3000 $Wm^{-2}°C^{-1}$ [26]. For comparison, this coefficient is between 25 and 500 $Wm^{-2}°C^{-1}$ for natural convection heat sinks [1].

We should underline, however, that including a fan in an electronic system will require an additional power supply. This will certainly impact the final size of the system.

4.1.2 Forced convection heatsink design

An important parameter in the design of a forced convection heatsink is the flow rate of the cooling fluid (e.g., air) going through the fins of the sink [2]. As a matter of fact, the flow rate generated by the fan actually impacts the heat transfer coefficient: A higher flow rate will mean a greater cooling fluid velocity, leading to a better convection heat transfer [1]. The flow rate also impacts the pressure drop across the heatsink given that for a given fin configuration, higher flow rates generate higher pressure drops[4].

Consequently, the proposed design procedure will search for *a balance between a reduced thermal resistance and an acceptable pressure drop*. It can be organized in the following seven-step procedure.

STEP 1: Collect heatsink pressure drop data.

Flow over a fanned heatsink generates a pressure drop that depends on several parameters including the number of fins, the distance between two fins, the value of the flow rate, etc. [26]. Suppliers of heat sinks will generally be able to provide, for the different models they propose, plots of the pressure drop versus the cooling fluid (air) flow rate. These plots are referred to as *heatsink working diagrams* where the flow rate is generally expressed in cubic feet per minute (CFM: ft^3/mn).

Figure 11 shows an example of such plots.

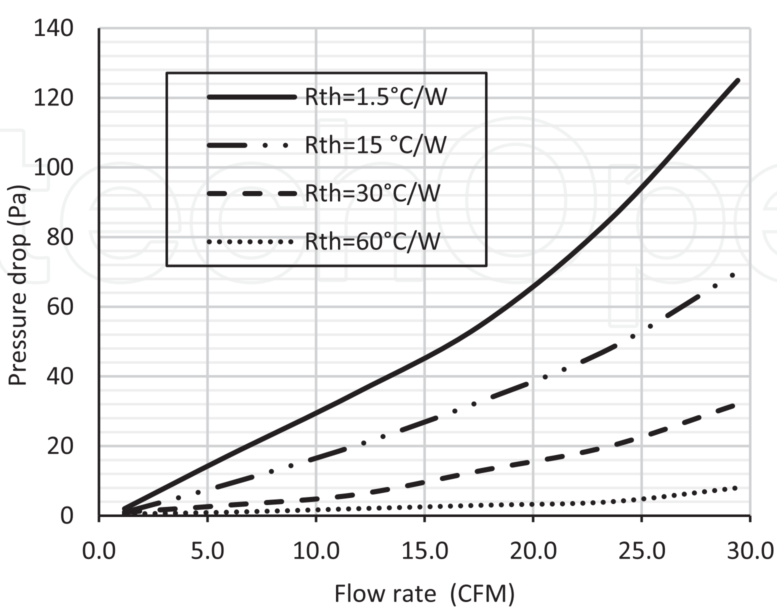

Figure 11.
Heatsink working diagram.

[4] For laminar flow, the pressure drop is proportional to the square of the flow velocity [26].

STEP 2: Get the fan curves.

Similarly, fan suppliers provide plots that give, for each model proposed, the flow rate achievable by the fan under a given pressure drop. These plots are often referred to as *performance curves of the fans*.

Figure 12 presents examples of performance curves for four fans.

STEP 3: Select heatsink candidates based on their working diagrams.

The selection criterion here is the individual thermal resistance, R_{th}^{s}.

STEP 4: Define the allowable coolant temperature rise.

The *coolant temperature rise* is defined as follows:

$$\Delta T = T_o - T_i \tag{17}$$

where T_i and T_o are, respectively, the inlet and outlet temperatures of the coolant.

T_i being known (generally, the coolant enters the heatsink around ambient temperature: $T_i \approx T_a$), the definition of ΔT permits to calculate T_o:

$$T_o = T_i + \Delta T \tag{18}$$

STEP 5: Determine coolant flow rate for each heatsink candidate.

For each heatsink s under evaluation, the thermal resistance, R_{th}^{s}, is known. This makes it possible to calculate the coolant flow rate by combining Eq. (9) and a heat balance on the cooling fluid:

Equation (9) gives:

$$Q_s = \frac{T_c - T_a}{R_{th}^{s}} \tag{19}$$

The heat balance for the cooling fluid gives the following:

$$Q_s = \dot{m}_s C_p (T_o - T_i) = F_s \rho C_p \Delta T \tag{20}$$

Where:

- $\dot{m}_s$ is the mass flow rate of the coolant fluid (air) over the heatsink s, kg/sec,
- C_p is its sensible heat (J/kg°C),

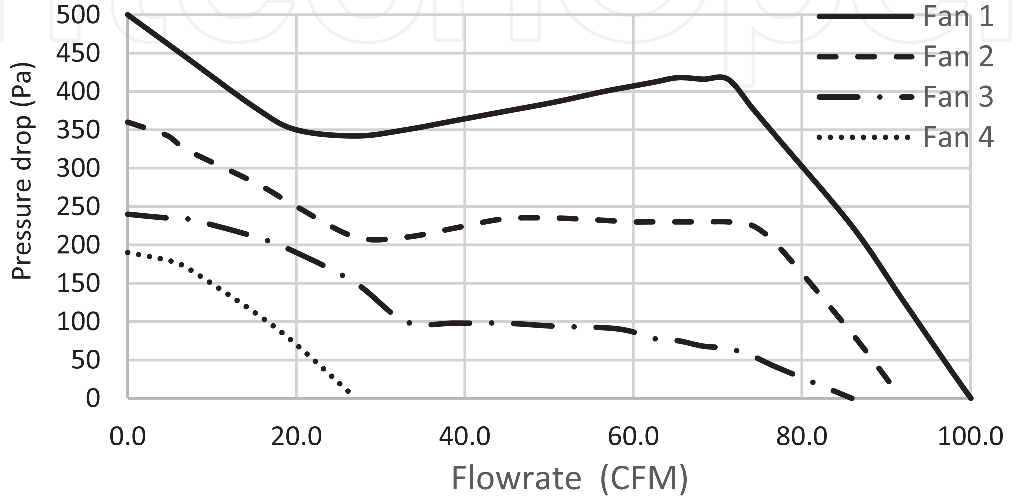

Figure 12.
Performance curves of fans.

- ρ is its density (kg/m^3), and
- F_s is the coolant volume flow rate (m^3/s).

Substituting for Q_s in Eq. (19) and extracting the coolant volume flow rate, F_s:

$$F_s = \frac{T_c - T_a}{\rho C_p \, \Delta T \, R_{th}^s} \tag{21}$$

STEP 6: Determine the pressure drop for each of the heatsink candidates.

Injecting the values of flow rates into the working diagram gives the pressure drop which will be generated by each of the heatsink candidates. **Figure 13** shows this procedure for the first two heat sinks considered in **Figure 10** (R_{th} = 1.5°C/W and R_{th} = 15°C/W), which permits the determination of pressure drops ΔP_1 and ΔP_2 generated by when F_1 and F_2 flow over sink 1.

STEP 7: Determine compatible fans using the performance curves.

For each heatsink candidate, s, inject the pressure drop, ΔP_s, determined in the previous step in the performance curves. This will generate the series of flows, $F_s^{fan_i}$, which will be delivered by fan i, operating against the pressure drop ΔP_s.

Figure 14 shows that, for fan 1, all performance curves cross the line ΔP_1. This means that any of the fans under evaluation can be used. The choice will then be made on price and volume criteria.

4.1.3 Cold plate heat exchangers

Cold plate heat sinks could be considered as a particular class of *plate heat exchangers* [27]. They are used when thermal powers released by electronic systems (smartphones, electric cars, onboard avionics systems, TGV, etc.) become so important that forced convection heat sinks are no longer sufficient [28–30]. They are constituted of plates that are fitted with pipes (see **Figure 15**) through which cooling fluid passes. They are attached to the surfaces of the electronic component or to the board to be cooled. The coolant is conveyed by means of a pump. The coolant itself is, in turn, cooled using a compact exchanger [27].

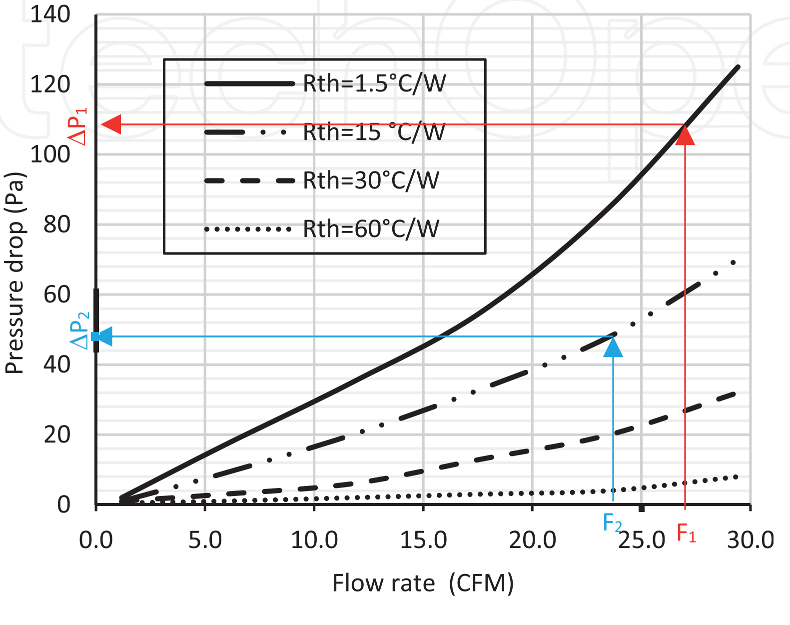

Figure 13.
Determination of pressure drops for heat sinks 1 and 2.

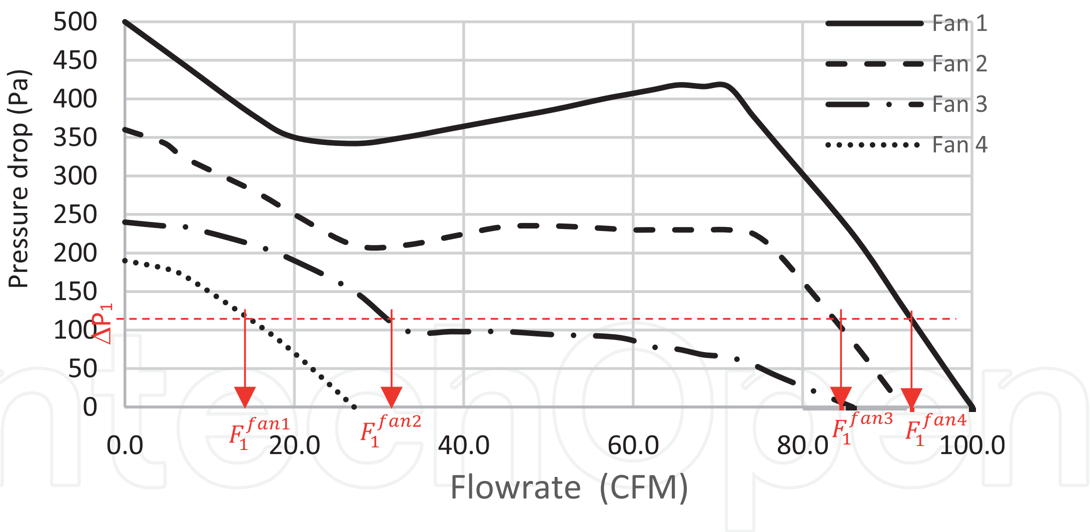

Figure 14.
Determination of deliverable flows under ΔP_1.

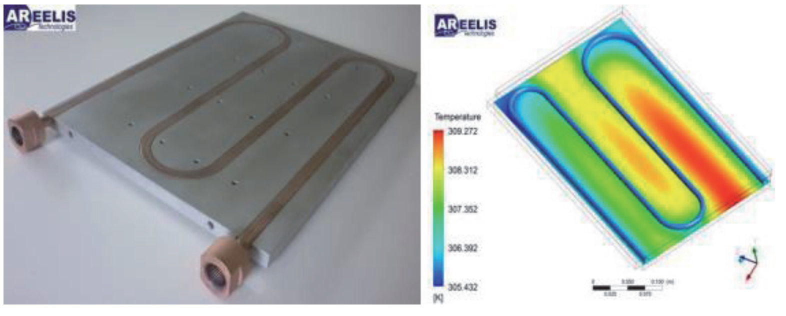

Figure 15.
A cold plate heatsink and its thermal scan. Sources [28, 29].

4.1.4 Microchannel heat exchangers

Microchannel heat sinks can be considered as a particular subclass of *printed circuit heat exchangers* [27]. These are indeed very small exchangers with overall dimensions not exceeding a few millimeters.

In contrast to their very small dimensions, they allow heat transfers in the order of 800 W/cm^2 [30–33]. The cooling fluid (generally air) circulates in microchannels, of microscopic equivalent diameters (approximately 10–60 μm), formed by etching on metal plates or in composite materials [34]. These microchannels have heights of the order of 0.5 mm. The modules thus formed are placed under the electronic components to be cooled [35]. **Figure 16** illustrates such an assembly where the circuit board to be cooled is shown in semi-transparency, above the microchannels.

However, microchannel cooling suffers from several drawbacks: complex implementation, significant pressure drops associated with microchannel flow. Moreover, it does not quite meet the requirements of thermal management in power electronic systems [36].

4.1.5 Refrigerated heat exchangers

Significant efforts have been devoted during the last 10 years to the development of a new kind of heat sinks. These are constituted of heat exchange plates associated with refrigeration cycles and a refrigerant as the heat transfer fluid [37–39]. Such refrigerated heat exchangers are used to cool power electronic

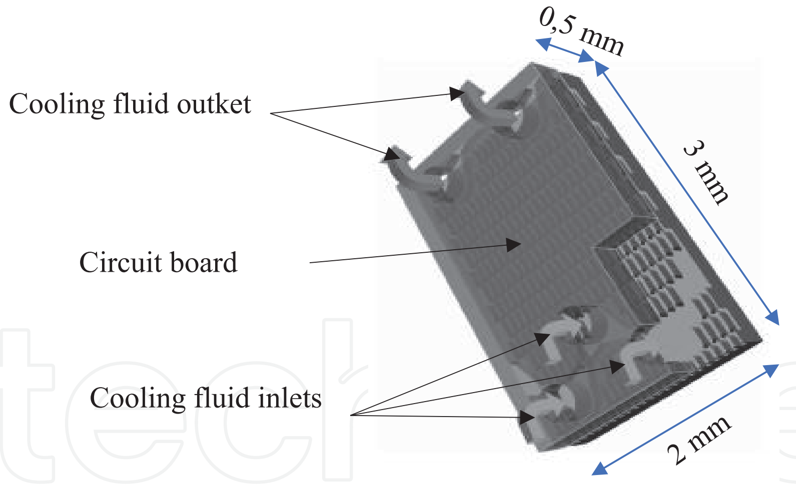

Figure 16.
Overall dimensions of a micro-channel heatsink.

systems such as laser power supplies and their optical systems. They are able to extract heat flux densities exceeding 1000 W/cm^2 while keeping microchips at temperatures below 65°C [39–44].

Note, however, that mounting this type of heat exchanger on electronic equipment remains difficult due to the fact that all the components of a refrigeration cycle (compressor, expansion valve, evaporator, and condenser) must be assembled in rather tiny spaces (**Figure 17**).

These systems must therefore meet an important challenge: the miniaturization required by designs favoring small sizes [21, 42, 43].

4.2 Two-phase heat sinks

These are phase change micro-exchangers, where the cooling fluid (generally a liquid) undergoes phase changes (from liquid to vapor and back to liquid) during the heat transfer process. Two-phase heat sinks are generally constituted of a vessel containing a heat transfer fluid; water in most cases. The lower surface of the vessel is in contact with the heat source (electronic equipment), and the upper surface is in contact with the cooling medium (generally air). **Figure 18** presents the operating principle of this system.

When in contact with the lower surface (heat source), the heat transfer fluid evaporates by absorbing the heat generated at this surface. The vapor then goes

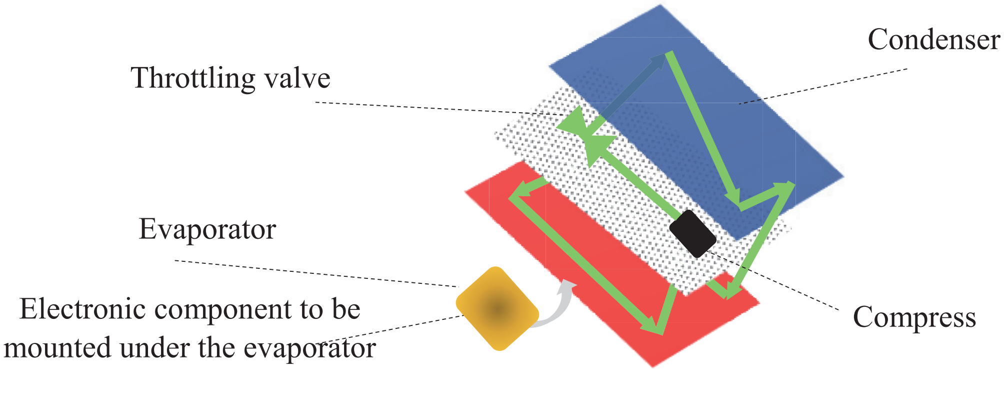

Figure 17.
Exploded view of a chip mounted under a refrigerated heatsink.

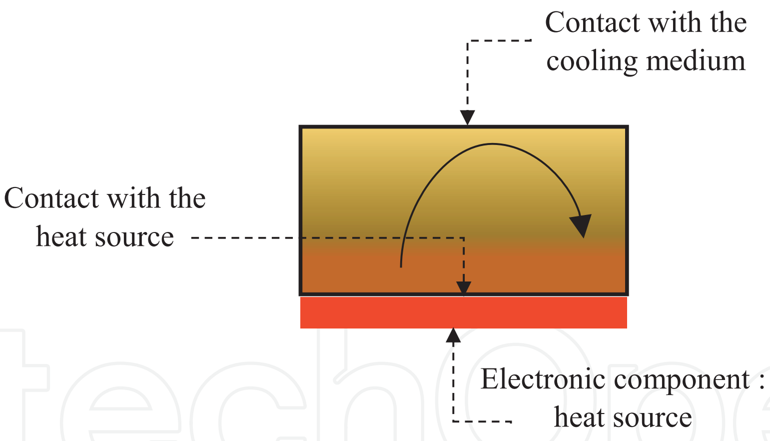

Figure 18.
Operating principle of a two-phase heatsink.

upwards, toward the upper surface where it gets cooled and condensates. Condensed liquid falls back along the walls of the vessel, down to the lower surface, where it absorbs again the heat and evaporates. This cycle is repeated as long as heat is generated by the electronic equipment.

This way, two-phase heat sinks are used to remove heat from the heat source to the ambient environment. The lower surface acts as an evaporator and the upper one plays the role of a condenser.

Note that two-phase heat exchangers involve latent heat instead of sensible heat in energy transfers, thus allowing large amounts of heat to be exchanged over small areas, which leads to great compactness.

Two-phase heat sinks include the following:

- Spray coolers
- Vapor chambers
- Heat pipes.

4.2.1 Spray coolers

Spray coolers are a special class of two-phase heat exchangers since the heat transfer fluid undergoes a series of evaporations and condensations [45, 46]. The electronic component is cooled by a jet of fluid, which partially evaporates while absorbing heat (see **Figure 19**). The heat transfer fluid is sprayed directly onto the surface of the power component to be cooled [47–49].

Spraying enhances the vaporization of the fluid even at relatively low temperatures (60–75°C, under 1 Atm.). It is generally realized using a single injector [50] or multiple injectors [51].

However, like refrigerated exchangers, the realization of spray cooling is relatively complex. As shown in **Figure 20**, it requires, in addition to the sprayers, the installation of a condenser to collect the vapors generated and a pump or compressor to feed the injectors.

4.2.2 Vapor chambers

Vapor chambers are the most common two-phase-heat exchangers where a heat transfer fluid (water in most cases) is placed in a sealed envelope: *the chamber* [52]. The lower surface of the chamber is mounted on the electronic component to be

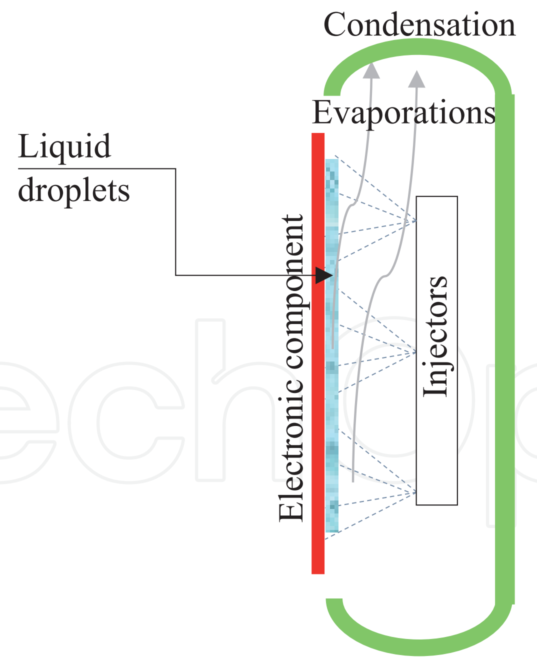

Figure 19.
Spray cooling system.

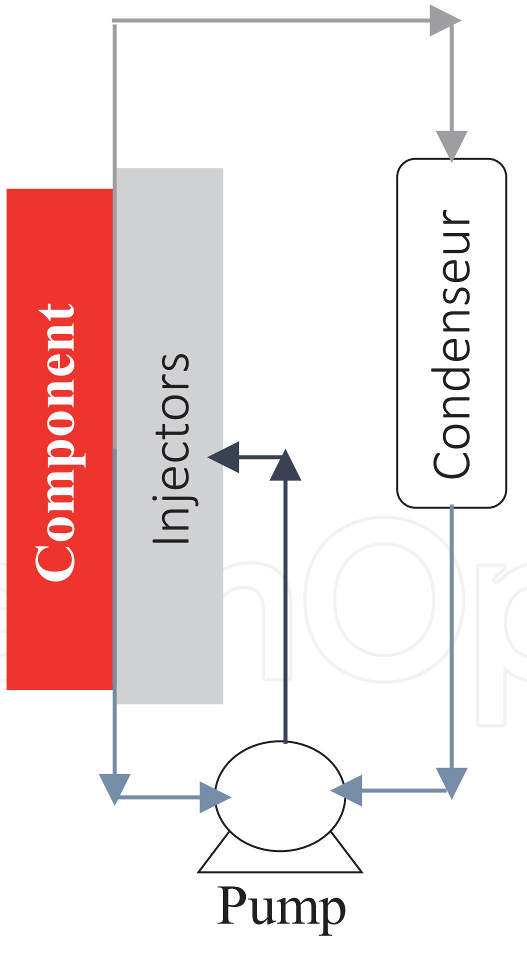

Figure 20.
Spray cooling cycle.

cooled. As described above, the liquid absorbs the heat generated by this component to evaporate. The vapors then condense on the cold surface of the chamber, which transfers the ambient environment the energy liberated by the condensation process [53, 54].

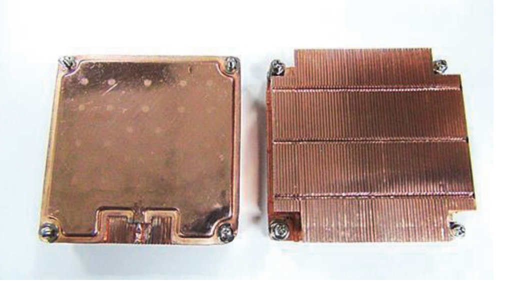

Figure 21.
Vapor chamber heat exchanger. Source [55].

Figure 21 shows this type of heat exchanger. It should be noted that the dimensions are extremely small to meet the requirements of miniaturization of onboard electronics and mobile telephony. The typical thickness of such an exchanger is 2–8 mm.

Current developments focus on the ultra-miniaturization of these exchangers by introducing ultra-thin evaporation chambers: thicknesses from 0.3 to 2 mm using walls made of titanium, stainless steel, or copper alloys. **Table 2** shows the current uses of these types of exchangers, as well as their typical dimensions and the thermal powers conveyed.

4.2.3 Heat pipes

A heat pipe consists of a sealed tube, containing a heat transfer fluid, without any other gas (see **Figure 22**). In most cases, only a small quantity of water suffices: About 1 cc of water for a 150 mm long, 6-mm heat pipe is typical.

In one of its zones (vaporization zone), the heat pipe tube is in contact with the hot source to be cooled. The heat recovered from this source increases the temperature of the heat transfer fluid causing it to evaporate.

The resulting vapors then accumulate in the condensation zone of the heat pipe where they condense on the internal walls of the tube, releasing their latent heat to the ambient environment (see **Figure 22**). The condensate flows in droplets on

Domain of use	Thickness (mm)	Length (mm)	Width (mm)	Power range (W)	Construction materials
Smart phones	0.3–0.4	80	50	5	Copper alloy or stainless steel
Portable microcomputers	0.5	220	60	15–25	Copper alloy
Base telecommunication stations	2–4	100	100	50–100	Copper or copper alloy
Graphic boards	2–4	200	90	200–350	Copper or titanium
Servers	3–4	150	80	200–400	Copper
Game consoles	3–5	150	150	150–200	Copper
Power inverters	3–8	450	450	500–2000	Copper

Table 2.
Vapor chamber uses and construction materials.

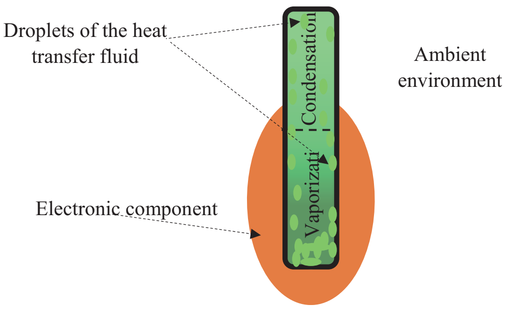

Figure 22.
Heat pipe working principle.

the walls of the tube and returns to the vaporization zone to be again submitted to the heat flow and to evaporate.

4.2.4 Heat pipe exchanger

These special heat exchangers are generally formed of a number of heat pipes usually made of copper, where several fins (the fin stack) have been mounted in the condensation zone.

As shown in **Figure 23**, each pipe is bent in a U form, slightly flattened where contact is made with the mounting base plate (hot area), and filled with a heat transfer fluid. When the copper pipes are going to be directly in contact with the environment, protection against corrosion could be obtained by nickel plating.

Each heat pipe acts actually as a *heat mover* from the area where heat is generated (the base plate) to the cooling area (fin stack) where air movement carries the heat away.

Rules of thumb

1. To maximize the amount of energy received from the heat source (the electronic component), each heat pipe should be placed directly above this source. The closer and tighter the contact is, the better is the heat transfer.

2. To improve the contact between the source and the heat pipe, sections of the pipes that are in contact with the source are flattened to about 30–65% of their original diameter.

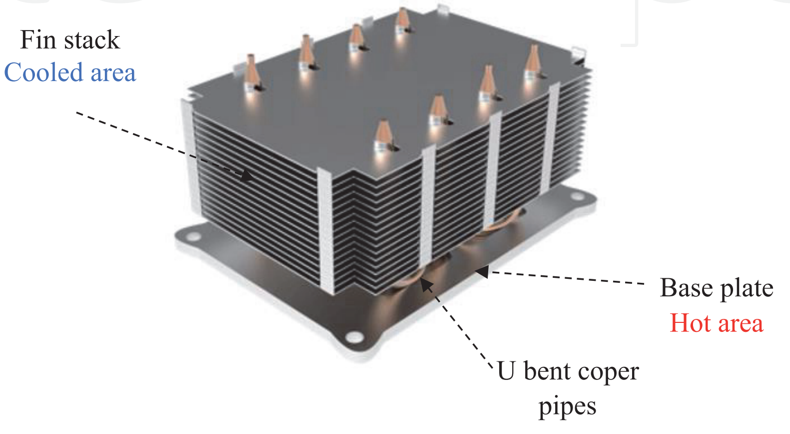

Figure 23.
Fixing the fin stack to the base plate.

3. The bend radius of the heat pipes is to be fixed with care in order not to alter the integrity of the pipes.

4. The minimum *bend radius* should be *three times the diameter of the heat pipe*.

Note that the heat pipes can be directly fixed on the electronic component to be cooled. **Figure 24** shows a heat exchanger made of six copper heat pipes that are directly fixed on a central processing unit (CPU) to convey heat to a 13 fin stacks. The advantage is a direct contact between the component and the heat pipes, thus reducing the thermal resistance of the base plate.

A more elaborated version is the U-shaped vapor chamber design where a U-shaped heat plate replaces the heat pipes, the heat plate operating in the same way as flattened pipes. **Figure 25** shows a heat exchanger formed from a vapor chamber bent to offer a large copper base to be in direct contact with the power electronic component. This design offers the advantage of a large direct contact area, which translates into a better performance: more than 20% better than the design of the pipe.

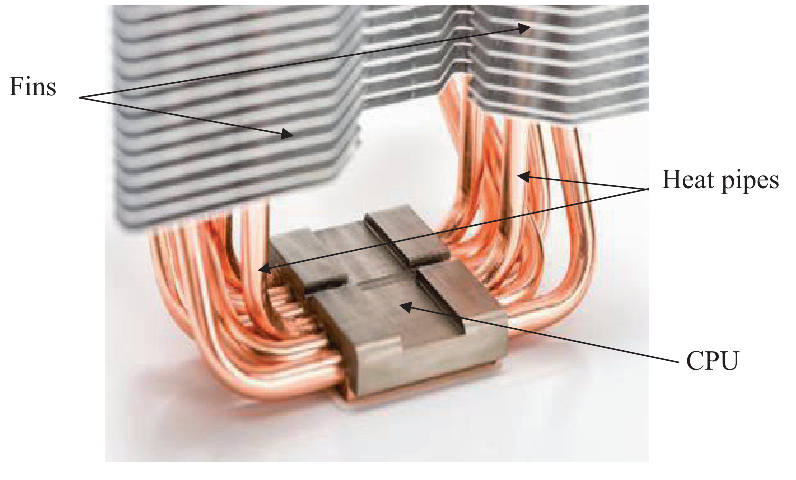

Figure 24.
Heat pipe heatsink to cool a CPU.

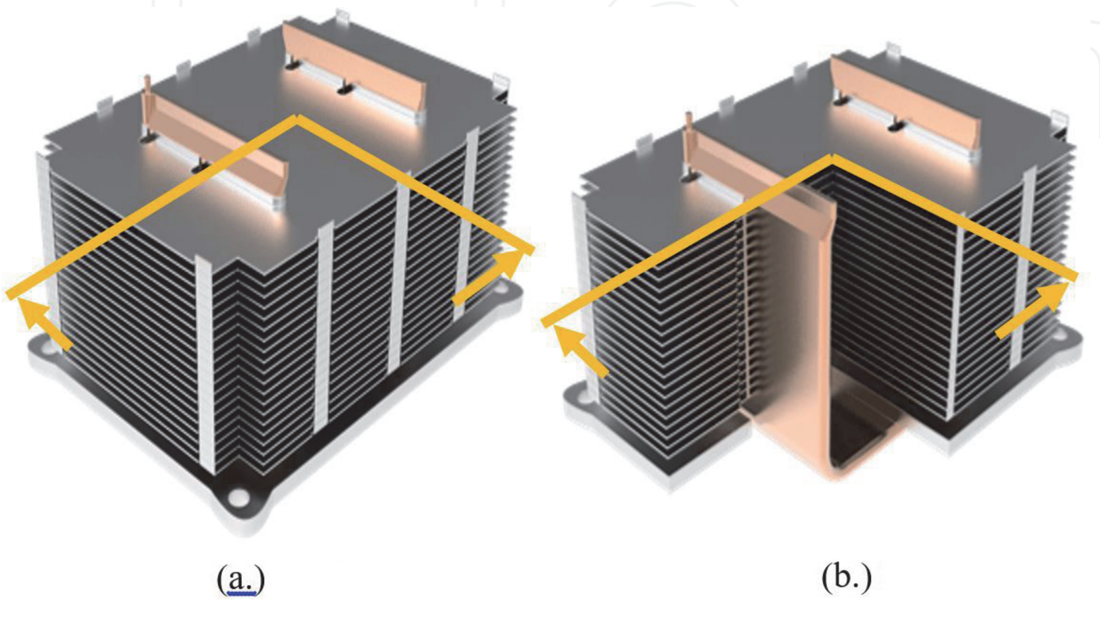

Figure 25.
U bent vapor chamber with a fin stack. (a) Complete vapor chamber and (b) showing inside the chamber.

4.3 Two-phase heat exchanger design

4.3.1 Design fundamentals

4.3.1.1 Carrying capacity

The performance of a two-phase heatsink is measured by its carrying capacity, Q, defined below.

Definition

The *carrying capacity*, Q, of a two-phase heat exchanger is defined as the amount of heat the device can move out per unit time.

From its definition, Q is an *energy per unit time*, which is a *power* which should be expressed in *Watts*.

For a given two-phase heat exchanger, the value of Q depends on several parameters: the size of the vessel, the latent heat of the transfer liquid used, the thermal conductivity of the metal constituting the vapor chamber or the heat pipe, convections (inside and outside the vessel), etc. However, we should note that, in two-phase heat sinks, energy is essentially transferred through the vaporization-condensation process, involving latent heat.

4.3.1.2 Two-phase thermal budget

Definition

The *thermal budget*, ΔT, of a two-phase heat exchanger is defined as the difference between the temperature of the electronic component (heat source), T_c, and the temperature of the ambiance where the component will operate, T_a:

$$\Delta T = T_c - T_a \tag{22}$$

The carrying capacity, Q, is proportional to the thermal budget ΔT (see Eq. (9)).

Thus for low thermal budgets, it will be difficult to achieve high values of Q. In this case, two-phase heat sinks should be used, mainly when $\Delta T < 40°C$.

4.3.1.3 Effective thermal conductivity

Heat exchange in two-phase sinks is actually complicated to model. It involves several heat transfer mechanisms: conduction through the vapor chamber or the heat pipe metal envelope, evaporation, condensation, convection inside and outside the vessel, etc. The exact representations of these transfers are interesting on the theoretical level of analysis. But, for sizing and design purposes, a more practical approach has been adopted by manufacturers and system designers. This approach is based on the thermal resistance model, in a way similar to that presented in Subsection 2.1 for single-phase heat sinks.

Let us recall that in conduction and in convection, thermal resistance models permit to express the power transferred Q in terms of R_{th} and the thermal budget, ΔT, as follows [2, 47]:

$Q = \frac{\Delta T}{R_{th}}$, where, R_{th} is the thermal resistance, which is expressed differently for conduction and convection:

- For a convection heat transfer:

$$R_{th}^{conv} = \frac{1}{hA} \tag{23}$$

Where h is the convective heat transfer coefficient and A the heat transfer area.

- For a conduction heat transfer:

$$R_{th}^{cond} = \frac{L}{kA} \tag{24}$$

Where L is the material thickness, A the heat transfer area, and k is the *thermal conductivity*.

Similarly, two-phase heat exchanger energy transfers are represented using an *effective thermal resistance*, R_{eff}, such that the carrying capacity, Q, is given by:

$$Q = \frac{\Delta T}{R_{eff}} \tag{25}$$

With $R_{eff} = \frac{L}{A\ k_{eff}}$, where k_{eff} is the *effective thermal conductivity* of the two-phase heat exchanger and A is its *heat transfer area of the heat sink*.

Definition

The *effective thermal conductivity*, k_{eff}, of a two-phase heat exchanger is defined as follows:

$$k_{eff} = \frac{QL}{A\Delta T} \tag{26}$$

Where Q is the *carrying capacity*, A is its *heat transfer area*, ΔT is the thermal budget (temperature difference between evaporator and condenser sections) and L is the *effective length* of the vessel which is the *distance from the midpoint of the evaporator to the midpoint of the condenser*.

It should be noted that, unlike the thermal conductivity of materials, the *two-phase heat exchanger thermal conductivity* varies with length. It is not a physical property of the material, but it is a modeling representation of heat transfer in two-phase heat sinks.

This modeling—using effective thermal conductivity, k_{eff}—makes it possible to represent *two-phase heatsink energy transfers* by *a simple thermal resistance model*, just the same way thermal resistance was used in single-phase heat sinks.

Effective thermal conductivities of two-phase heat exchangers range from 1500 to 50,000 W/m°C. For comparison, the thermal conductivities of the best energy conductors are around 400 W/m°C. Effective thermal conductivities of two-phase heat exchangers can reach 10–100 times the conductivity of the best thermal conductors like copper.

4.3.2 Design steps

Generally, manufacturers of two-phase heat exchangers supply, among other information on their product, the following parameters:

- Operating temperature limits: generally between 20 and 250°C.
- The heat exchanger carrying capacity.
- The heat exchanger thermal resistance.

The following five-step procedure is proposed to design two-phase heat exchangers.

STEP 1: The datasheet of the electronic component to be cooled gives the power and operating temperature: Q (W) and T_{op} (°C).

STEP 2: Knowing ambient temperature, T_a, calculate the thermal budget ΔT as follows:

$$\Delta T = T_{op} - T_a \tag{27}$$

STEP 3: Calculate the *design thermal budget* ΔT_D as follows:

$$\Delta T_D = \Delta T - 5 \tag{28}$$

STEP 4: Divide the *design thermal budget*, ΔT_D, by the power, Q, of the electronic equipment to be cooled. This division will give the maximal thermal resistance to be assured by the two-phase heatsink: R_{Max}

STEP 5: Select the two-phase heatsink having a thermal resistance $R < R_{Max}$.

Rules of thumb

1. Derate the thermal budget by 5°.

2. In the case of heat pipes, if bending is considered, derate the carrying power by 2.5% for every 45-degree bend.

3. When heat pipes are flattened to ensure a better contact with the base plate, the flattening should be in the range of 30–65%.

4. Flattening implies a 15–30% reduction in the carrying capacity of the heatsink.

A. Appendix 1

Determination of heat transfer coefficients for electronic components and circuit boards under natural convection

A.1 Electronic components under natural convection in air

$$\mathrm{h} = 3.53\left(\frac{\mathrm{T_w} - \mathrm{T_a}}{\mathrm{H}}\right)^{0.25} \tag{29}$$

Where:
H is the height of the small electronic component, *expressed in meters*
T_w and T_a are, respectively, the wall temperature and the ambient temperature

A.2 Circuit boards under natural convection in air

$$\mathrm{h} = 2.44\left(\frac{\mathrm{T_w} - \mathrm{T_a}}{\mathrm{H}}\right)^{0.25} \tag{30}$$

Where:
H is the height of the electronic board considered, *expressed in meters*
T_w and T_a are, respectively, the board and the ambient temperatures.

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