# Qpedia Thermal eMagazine

Volume I, Issues 1-12

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Book Excerpt

# *Qpedia* Thermal eMagazine

## - Volume I, Issues 1-12

Book Excerpt

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Advanced Thermal Solutions, Inc.

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**ISBN - 978-0-615-23660-5**

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*Cover Design & Pre-Press:* Sheling Wang

*Printed in China*

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## *Dedication*

To the hard-working and

professional staff of

Advanced Thermal Solutions, Inc.

Thank you for

all of your work

in developing Qpedia.

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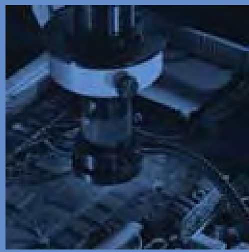
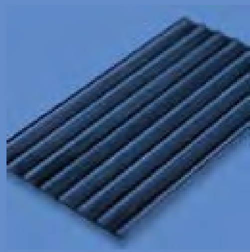

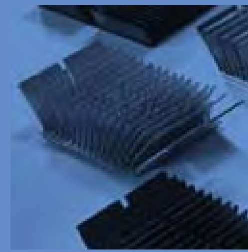

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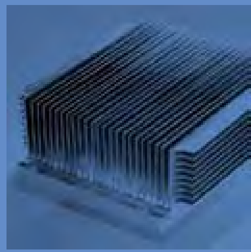

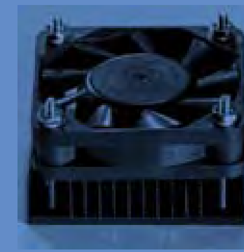


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# Jet Impingement Cooling

Continued power increases in electronic devices, such as processors IGBTs are requiring high capacity cooling techniques to remove excess heat. One such method is jet impingement of liquid or gas onto a surface on a continuous basis, if the noise issue can be addressed. This mode of heat transfer has been tested extensively for many decades and understanding its intricate dynamics is still an ongoing activity. The coolant for jets can be either air or some form of liquid (generally water).

There are three jet configurations: free-surface, submerged and confined. Figures 1a and 1b show a confined jet. Thermal transport of jet impingement has one of the highest heat transfer coefficients as compared to other cooling techniques. Reference [1] shows that a submerged jet has higher heat transfer coefficient than the free-surface jet for Reynolds (Re) numbers greater than 4000. In [2] it is shown that the confining wall can reduce the heat transfer coefficient due to the circulation regions between the top plate and the bottom surface.

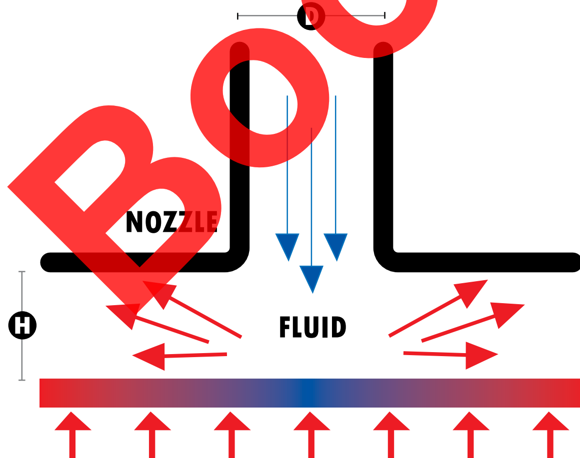


**Figure 1a. Schematic of a confined jet.**

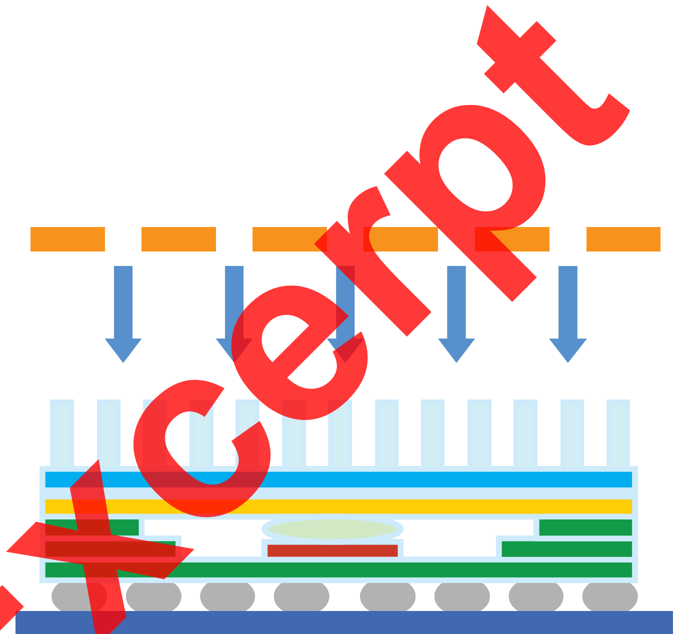

**Figure 1b. Multi-jet cooling for high power BGA applications.**

The heat transfer process and fluid dynamics in the jet impingement mode are very complicated and depends on many factors, but the main factors are the Reynolds and Prandtle (Pr) numbers, jet diameter and wall-to-nozzle spacing, H/D. It has been shown that for the same Reynolds number, decreasing jet diameter increases the heat transfer coefficient due to the coolant's higher speed. For a constant diameter jet, the heat transfer coefficient is a function of $R_e^{0.8}$. For certain values of jet distance to jet diameter, reducing the distance does not make an appreciable difference in the heat transfer, due to the potential core being so close to the surface.

Jets are deployed in different shapes, including round or square, that impact the eventual heat transfer from the impinged surface. There can be a single jet or many smaller jets, where the data has shown to have a better performance than a single jet. Figure 2 shows the temperature gradient in a jet impingement application [3]. The jet was seeded with liquid crystal to show the thermal transport, its interaction and temperature gradient.

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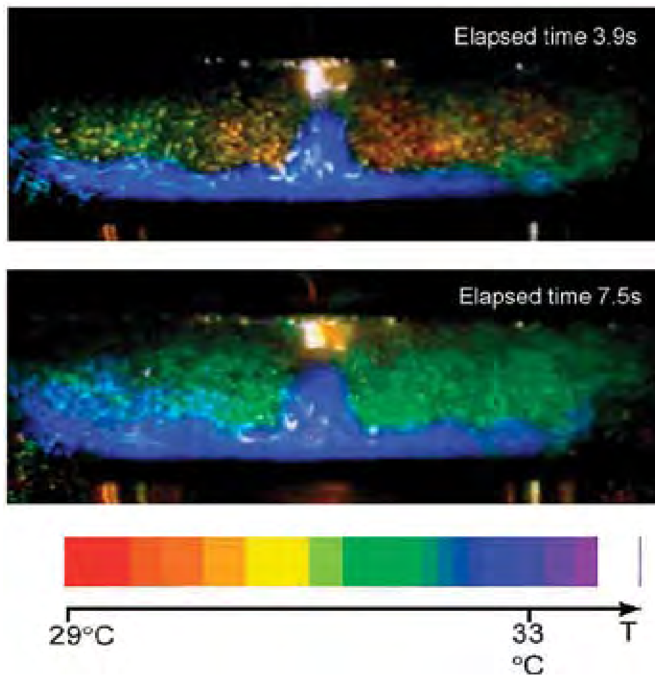


**Figure 2. Temperature gradient in a jet.**

The heat transfer coefficient is maximized at the center and its distribution is shown in Figure 3 [4].

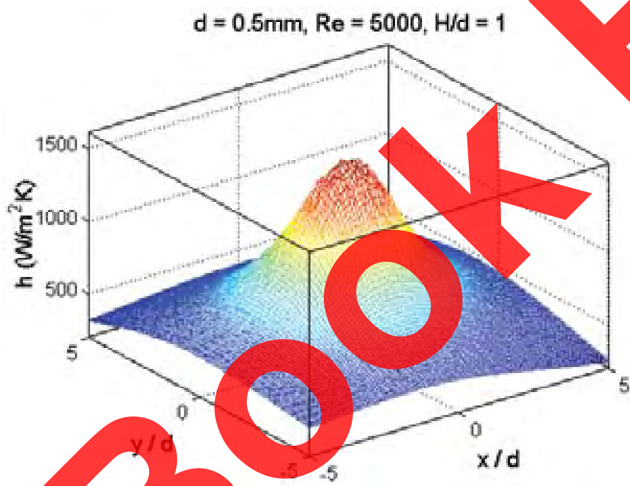


**Figure 3. Heat transfer coecient distribution [4].**

Figure 4 [4] shows the local heat transfer coefficient for air jet impingement as a function of dimensionless distance for a jet diameter of 1 mm, Re=10,000 at different H/D. It is seen that at H/D = 1, h is higher than the other two values. In this experiment an h=2,500 W/m².K was achieved.

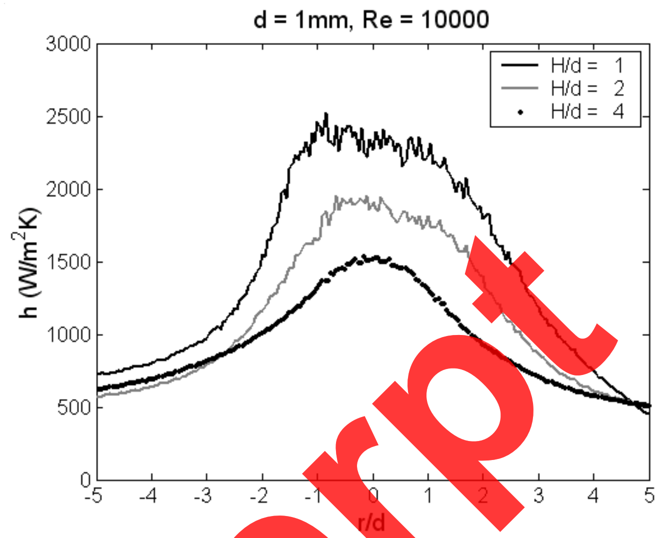


**Figure 4. Heat transfer coecient for an air jet at d=1mm, Re = 10000 [4].**

Figure 5 shows the heat transfer coefficient for a water jet reported by Garimella which has attained a value of 60,000 W/m².K for confined submerged jets. The x axis is the ratio of jet-to-target-spacing to nozzle diameter. The top curve was for 10 gram/sec (4.6 m/sec), the middle curve was for 6.9 m/sec and the bottom was for 9.2 m/sec.

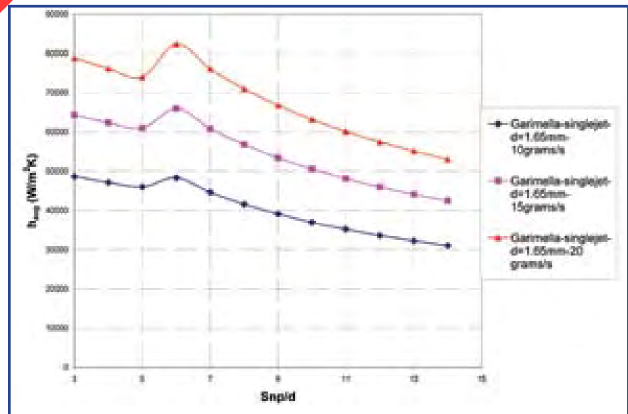


**Figure 5. Heat transfer coecient for single submerged conned water jet [2].**

Recently, Motakef, et. al. [5], have shown that they can achieve a heat transfer coefficient of 500,000 W/m².K for a water microjet and 20,000 for an air microjet. In their design, they have manufactured a 3-D structure with hundreds of microjets with the size of 300 microns. The jet is kept at a distance of a few hundred microns from the sur-

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face. The special manifold design lets the returned flow to exhaust without interfering with the main jets, hence significantly increasing the heat transfer coefficient. Without the manifold, the heat transfer coefficient degrades to a macro jet. Figure 6 shows a sample of this honey-comb structure.

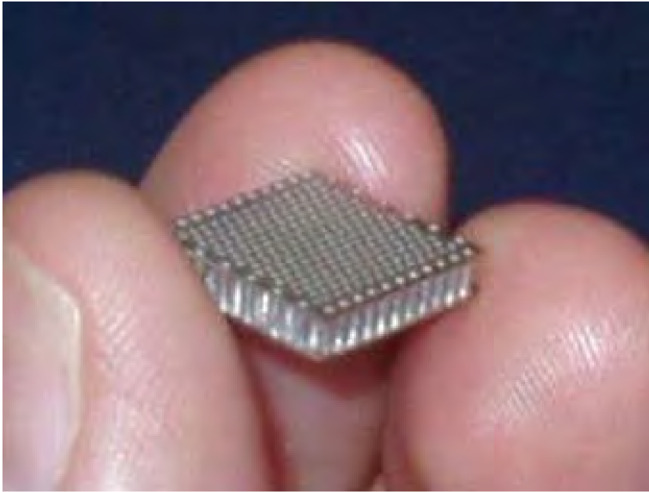

**Figure 6. A 10x20x1.7 mm MJCA microjet structure.**

Even though a very high heat transfer coefficient can be achieved using jet impingement, the packaging of such a system is very challenging. The following must be carefully considered and studied:

1. What type of fluid should be used? Is it air or liquid? If it is air, what is the noise implication? If it is liquid, how will the liquid be drawn out of the system without damaging the electronic parts if there is leakage?

2. What type of compressor is needed to generate the high speed jet? Is its size practical for the commercial use? What is its life span?

3. What type of filter should be used to prevent the nozzle from clogging? What is the effect of the filter on the fluid line pressure drop? What would be the implication of semi-clogged nozzles on the pressure drop?

4. What is the cost of such a system and does it justify its use for that specific application?

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# Thermal Management of Telecom and Datacom Equipment

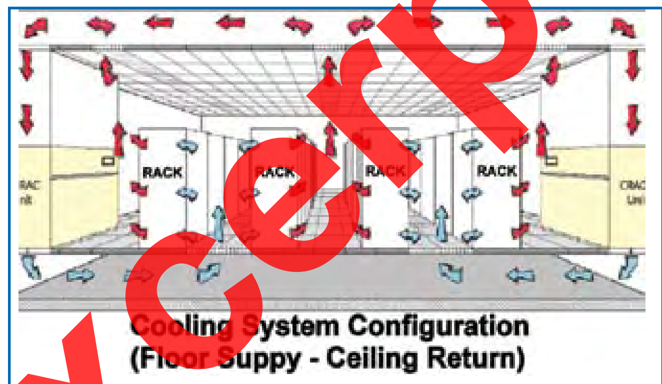
Because of their unique product offerings, communications and computing industries drive a large segment of the technology market. The power dissipation of equipment doing faster data transmission and processing is on the unwavering rise. For that reason, proper thermal management of this gear has a central role in its successful deployment and operation.

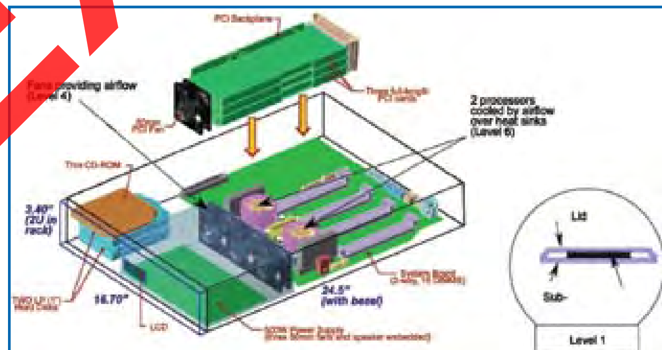
End-users insist that their telecom and datacom equipment meet compliance and performance standards. For the manufacturers, one of the most difficult requirements to satisfy is effective thermal management.

The basis of this thermal challenge resides in several parameters:

- Thermal coupling within the system and surrounding equipment
- System standardization (e.g., the ATCA Standard)
- Constrained space
- EMI/EMC requirements for high frequency devices, boards and packaging
- Acoustic noise
- Limited air flow
- Rigid performance standards (e.g., 72 hours operation at 55˚C)
- Non-uniform power distribution and congested PCBs
- Field serviceability (e.g., system operational within 4 hours of failure)

A successful thermal design requires that the junction temperatures of all critical devices in a system be sufficiently below their critical level in the worst-case ambient temperature. Otherwise, higher temperatures may result in data transmission bit errors and/or a decrease in a system's life expectancy. Thermal coupling and non-uniform PCB layout make this determination a complicated process. Figure 1 shows an example of thermal coupling from environment to the device



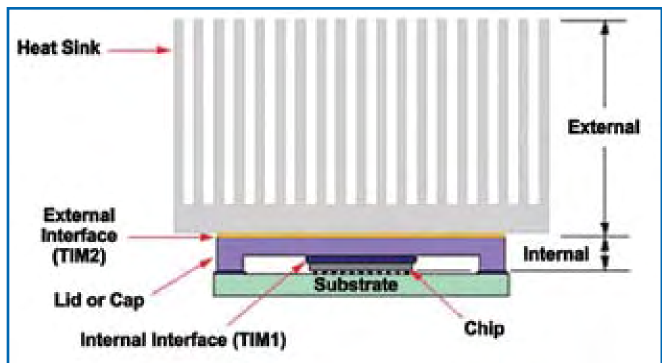




**Figure 1. Thermal coupling in telecom and datacom equipment, from the central oce (data center) to the component.**

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**Table 1. Solution hierarchy for thermal analysis.**

| Solution Order | Package Level | Expected Results | Requirements and Tools |
|---|---|---|---|
| First | System | System flow distribution and boundary conditions for the card racks, T, V, and Pressure | Measurement or Simulation (CFD) |
| Second | Card Rack (chassis) | Boundary conditions for the PCB, and thermal coupling between the board and the rack | Measurement or Simulation (CFD) and solid modeling (temperature gradient in the solid) |
| Third | Board (PCB) | Boundary conditions for the component on heat transfer and fluid flow | Measurement of fluid flow distribution and board properties and board level solid modeling |
| Fourth | Component (device or module) | Boundary conditions for the Die ($T_j$) and its cooling solution | Fluid flow and temperature measurement equipment; Heat sink; Interface materials and their properties; Solid modeling; Package material properties |

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In telecom and datacom equipment, to calculate the junction temperature of a device residing on a PCB, all parameters impacting its magnitude must be accounted for. See Figure 2.

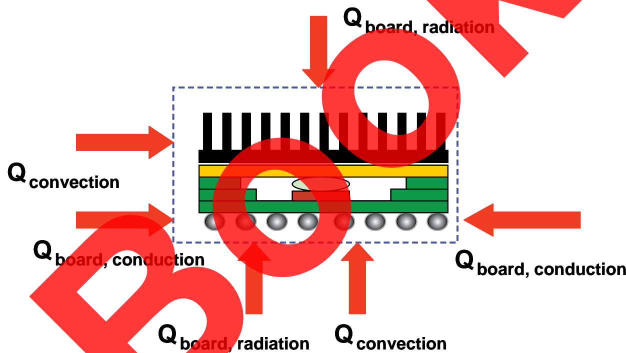


**Figure 2. Thermal coupling at the device level.**

The junction temperature ($T_j$) is a function of following parameters:

$$T_j = f(V_{fluid}, k_{board}, k_{component}, k_{spreader}, k_{interface\ material}, h_{heat\ sink}, T_a, P_{device}, \varepsilon, Q_{ustream})$$

Where V is the air velocity, k is the thermal conductivity, and ε is the emissivity.

Table 1 shows the approach, domain, anticipated results, and the needed tools to successfully obtain the junction temperature of a device in the system.

Once the junction temperature is obtained by using two independent methods, it must be ensured that there is at least a 10% margin of safety in the design. Therefore, the following equation must be satisfied:

$$\eta = (T_{j,\ calculated} - T_{a,\ reference})/(T_{j,\ spec} - T_{a,\ reference}) \leq 90$$

Where

$T_{j,\ calculated}$ = junction temperature as the result of above calculation.

$T_{j,\ spec}$ = critical junction temperature specified by the manufacturer.

$T_{a,\ reference}$ = reference ambient or approach air temperature.



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With a properly determined junction temperature in hand, a cooling method can be selected for maintaining the desired performance. Depending on the data obtained from Table 1, we can choose from these options:

- Natural convection
- Forced convection (air-mover)
- Active air cooling (not very common)
- Jet impingement (air or liquid)
- Advanced systems (liquid or refrigeration)

Although cooling by air is the most common and preferred system, liquid cooling has been used in unique circumstances. Regardless of the method selected, every cooling decision should consider the following parameters:

- **Cooling capacity** – Does it satisfy the junction temperature requirements?
- **Size** – Does it comply with the packaging requirements?
- **Regulatory requirements**– Does it meet the system and site implementation requirements (e.g., NEBS)?
- **Reliability** – Does it meet the expected life requirements?
- **Budget**– Does it comply with the cost constraints imposed on the system?
- **Market availability** – Is it readily available? Can supply-chain requirements be met?

Obtaining a successful cooling solution is made possible by combining methodical thermal analysis (as delineated in Table 1) with the results from at least two independent approaches, and by taking into account the above system parameters.

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# Pressure Drop Calculations in a Chassis

Before performing thermal analysis on a chassis, it is necessary to calculate total pressure drop within the system. This consists of the pressure drops of all system subcomponents. The total pressure drop (system curve), which also depends on the volumetric flow rate in the system, can be shown as:

$$H_{system} = R_{total} \times G^2\text{, where, } R_{total} = \sum R_i \qquad (1)$$

Where $R_i$ are the system component resistances, and G is the volumetric flow rate.

Generally, one or two fan trays can be used to deliver air through a chassis. The operating point of a fan tray is where the system curve intersects the fan tray curve.

Figure 1 shows the operating point. It is recommended to operate the fans below the stall range.

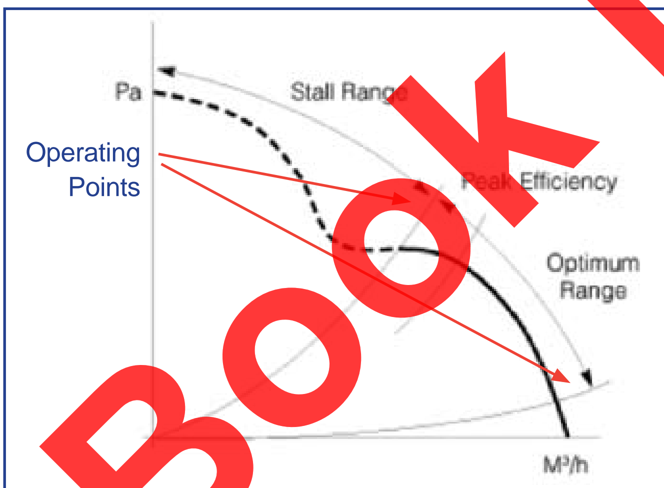


**Figure 1. Graph showing the intersection of system and fan curve.**

This intersection equals the system pressure, and thus provides the flow rate through the system. Total pressure drop is generally affected by the following subcomponents if they are part of that specific chassis:

1. Air filter
2. Honeycomb
3. Fan tray due to sudden expansion and contraction of flow through blades
4. Flow bending from inlet to plenum
5. Contraction to the inlet plenum
6. Expansion from inlet to plenum
7. Circuit cards component blockade
8. Circuit card friction (generally small)
9. Flow bending to exit plenum
10. Contraction to exit of plenum
11. Expansion from exit of the plenum

The following shows the calculation of several of the above resistances. The units are square meter for the area, cubic meter per second for volumetric flow rate and Pascal for pressure.

A. Perforated plate:

$$R = \frac{0.828}{A^2}\ \text{Pa/(m}^3\text{/sec)}^2\text{)} \qquad (2)$$

Where A is the area of open holes exposed to air ow

B. Filter:

$$R = \frac{L \times 510.79}{A^2} \qquad (3)$$

Where A is the lter exposed area and L is the lter loss coefficient provided by manufacturer.

C. Boards:

$$R = \frac{4.2 \times L}{A^2} \qquad (3)$$

Where L is the board length and A is the effective free area of the channel. If there are N similar boards mounted in a chassis in parallel, their equivilent resistence is calculated as:

$$\frac{1}{\sqrt{R_{combined}}} = \frac{N}{\sqrt{R_{board}}} \qquad (4)$$

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D. Fan trays contraction and expansion:

The flow going into and out of the fan trays exhibits contraction and expansion with their associated losses.

Sudden Expansion:

$$R = 0.46 \times \left[ \frac{1}{A^1} \times \left( 1 - \frac{A^1}{A^2} \right) \right]^2 \quad (5)$$

Where $A_1$ is the small area and $A_2$ is the larger.

Contraction:

$$R = \frac{0.321}{A^2} \quad (6)$$

Where A is the small area

E. Fan tray

When fans are mounted in a fan tray, the combined fan curve is calculated based on the fan laws. For M fans in parallel assembled in one fan tray, the fan volumetric flow rate increases by a factor of M for the same pressure drop. For two fan trays in series such as a push-pull system, pressure doubles for the same volumetric flow rate. So if the characteristic curve of one fan is:

$$\Delta P = f(G) \quad (7)$$

The characteristic curve of the combined fans can be represented as:

$$\Delta P = 2f(MG) \quad (8)$$

The factor 2 is for two fan trays and M is the number of fans in one tray.

The total head generated by a push-pull system is:

$$\Delta P = 2f(MG) + 0.658 \times \left( \frac{G}{Af} \right)^2 \quad (9)$$

Where is the fan duct size.

The second term is the velocity pressure head associated with the entrance fan that contributes to the total pressure. Solving the following equation, it would yield the system pressure drop and volumetric flow rate:

$$\Delta P = H_{system}$$

The above relations can be used for a quick analysis of a system to find the volumetric flow rate, which can then yield the system temperature rise if the total heat dissipation is known. As a simple example, consider a system that has two fan trays. Each fan tray has two fans. This system has five perforated plates with a 56% opening. Let's assume the area open to air flow is about 0.03 $m^2$ and that the system also has a filter at the inlet with a loss coefficient of 0.008. The fan curve indicates that maximum pressure is 225 Pa at zero flow and 0.13 $m^3$/sec at zero pressure. Assuming a linear relationship between the pressure and the volumetric flow rate would give the single fan curve as:

$$\Delta P = 225\text{-}1666G \quad (10)$$

Assuming negligible resistance from boards and fan hubs, Table 1 shows the resistances due to all the parameters and their calculated values.

Solve the following equation:

$$19208G^2 = 2\left(225 - \frac{1665}{2}G\right) \quad (11)$$

Which yields a value for G equal to 0.12 $m^3$/sec for the system volumetric flow rate.

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**Table 1. Flow resistances for dierent system components.**

| Component | Value |
|---|---|
| Inlet Cross section ($m^2$) | 0.03 |
| Single perforated plate resistance | 2933 |
| Combined perforated plate resistance | 14668 |
| Filter resistance | 4540 |
| Total system resistance | 19208 |
| Fan curve | 225-1665XG |
| Push pull Fan tray curve | 2(225-1665/2*G) |

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# Thermocouples: What They Are and How They Work

With their versatility and ease of use, thermocouples are the most common means for temperature measurement. Thermocouples date back to 1821, when Thomas Seebeck, an Estonian scientist, found that when two dissimilar metals with P/N characteristics were connected at both ends, and one end was heated, there was a resulting flow of current in this thermoelectric circuit.

Today, this voltage is known as electromotive force (emf), and is a function of the junction temperature and composition of the two metals. The repeatable nature of emf, along with standardized material types and calibration curves, have made thermocouples the devices of choice for many temperature measuring applications.

Despite the simplicity of thermocouples, their emf voltage cannot be measured directly. As shown in Figure 1, using a voltage measuring device will create two additional thermocouple junctions, designated as $J_2$ and $J_3$ in the thermoelectric circuit.

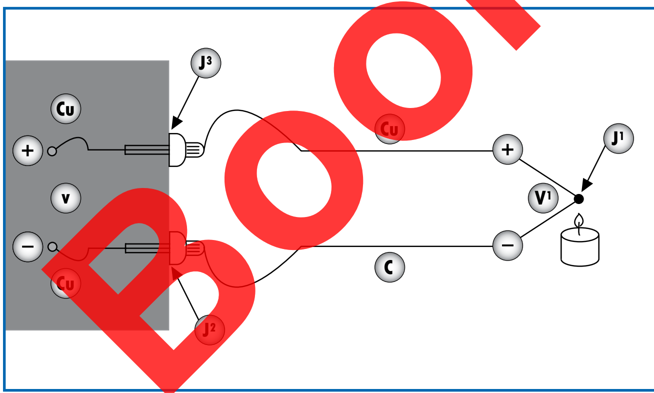


**Figure 1. Measuring junction voltage with a DMV [1].**

The voltage reading then becomes a function of three temperatures, two of which are of no interest to the experimenter. To solve this problem, another junction needs to be created in series in the circuit, as shown in Figure 2.

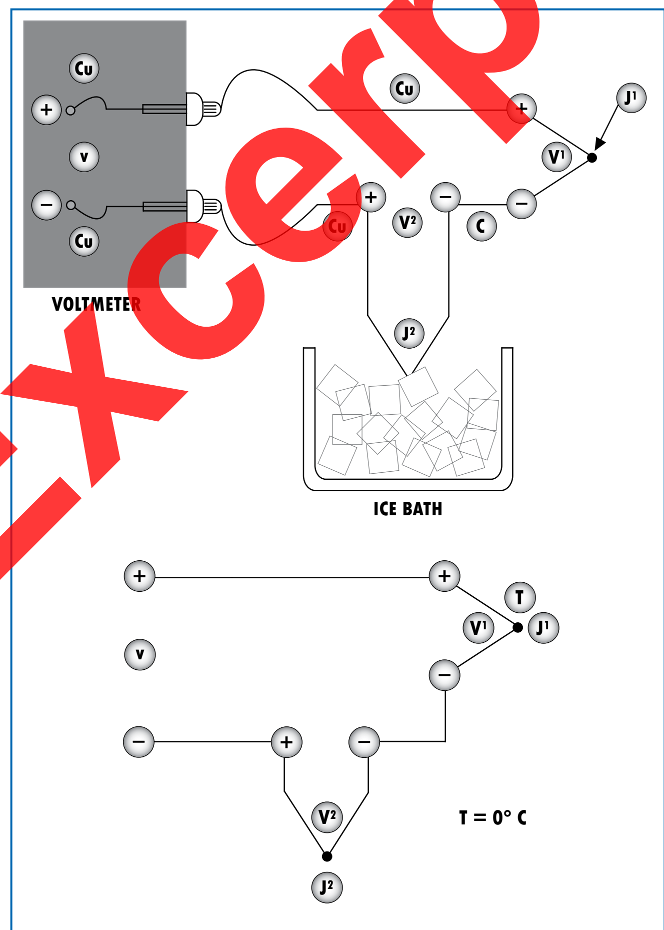


**Figure 2. External reference junction [1].**

This added junction is kept at a known temperature, 0°C, by submerging it in an ice bath, and thus becoming the reference temperature. Most commonly, a mixture of pure water and pure ice at 1 atmosphere of pressure is used for the reference junction. Today, electronic measuring devices simulate an ice reference bath without the need for an actual ice bath. This is often referred to as cold junction compensation (CJC).

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**Table 1. Different types of thermocouples.**

| Thermocouple type | Material Composition | Temperature Range °C | Uncertainty | Color Code |
|---|---|---|---|---|
| t | Cu vs. Constantan | -250 to 350 C | Greater of 1C or 0.75% | Blue-Red |
| k | Chromel vs. Alumel | -250 to 1250 C | Greater of 2.2 C or 0.75% | Yellow-Red |
| j | Iron vs Constantan | 0 ti 750 C | Greater of 2.2 C or 0.75% | White-Red |
| r | Platinum vs. Platinum-13% Rodium | 0 to 1450 C | Greater of 1.5 C or 0.25% | None Established |
| s | Platinum vs. Platinum-10% Rodium | 0 to 1450 C | Greater of 1.5 C or 0.25% | None Established |
| c | Tungsten 5% Rhenium vs. Tungsten 26% Rhenium | 0 to 2320 C | Greater of 4.5 C to 425C, 1% to 2320 C | None Established |
| e | Chromel vs. Constantan | -200 to 900 C | Greater of 1.7 C or 0.5% | Purple-Red |

After adding the ice bath, there are still two junctions at the device terminals, but each of these consists of the same two materials. If held at the same temperature, their emf voltage will be the same and cancel each other out.

Thermocouples can be made of any two dissimilar metal wires that can create a PN junction, and their emf voltage depends on the composition of the chosen metals. However, what makes thermocouples so popular is that the materials used to construct them are restricted and their outputs emf have been standardized. Certain materials and combinations are better than the others, and some have basically become the standard for given temperature ranges. Table 1 lists some of the available thermocouples on the market.

Selecting the right type of thermocouple for an application depends on many factors. These include sensitivity, temperature range, corrosion resistance, linearity of output voltage, and cost. For example, types R and S are relatively expensive and are not sensitive. However, they perform well at high temperatures up to 1768°C and are resistant to a number of corrosives. Type C thermocouples are suitable for higher temperature applications, but they are relatively expensive and corrode easily in an oxidizing environment. A & T type thermocouples are inexpensive and very sensitive but will corrode at temperatures above 400°C. Type K is very popular for general use, relatively inexpensive, reasonably corrosion-resistant, and can be used at high temperatures, up to 1372°C. K-type thermocouples also provide relatively linear output as compared to the other types [2].

The actual magnitude of the thermocouple emf is very small, and is in the order of few millivolts. At a given temperature, Type E has the highest output emf among common types, but this voltage is still measured in millivolts. The sensitivity of thermocouples is also relatively low. For instance, the voltage change per degree Celcius from 38 to 93°C is only 36 microvolts. As a result, thermocouples require accurate and sensitive measuring devices and cannot be used for temperature changes of less than 0.1°C. Traditionally, expensive voltage balancing potentiometers were used to measure emf. Today, a high quality digital voltmeter is sufficient [3].

The National Institute of Standards and Technology (NIST) has developed standard calibration curves for determining temperature based on the measured emf voltage. These data represent the output emf of thermocouples when an ice cold junction is used, and are incorporated in the memory of most DAQ systems. Unfortunately, the temperature-voltage relationship of thermocouples is nonlinear and curve-fitted using polynomial functions. Obviously, the higher the order of the polynomial function, the higher the accuracy of temperature reading. The polynomial function should only be used inside the temperature range of the thermocouple type and should not be extrapolated. To save computational time, a lower order polynomial fit can be used for a smaller temperature range.

Thermocouple wires come in a variety of sizes. Usually, the higher the temperature, the heavier the wire should be . As the size increases, however, the time response to

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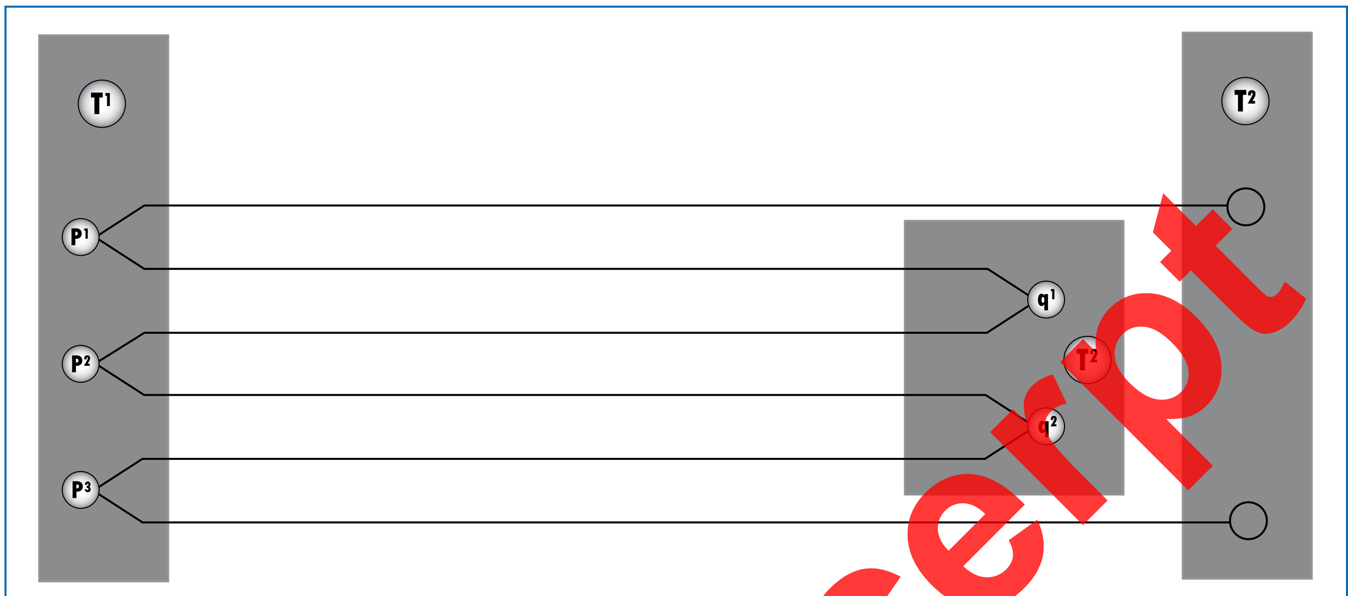


**Figure 3. Series-connected thermocouples forming a thermopile.**

temperature change increases. Therefore, some compromise between response and life may be required.

Thermocouples can be connected electrically in series or in parallel. When connected in series, the combination is usually called a thermopile, whereas there is no particular name for thermocouples connected in parallel. A wiring schematic of a thermopile combination is shown in Figure 3

The total output from n thermocouples will be equal to sum of the individual emf's. The main purpose of using a thermopile rather than a single thermocouple is to obtain a more sensitive element. Parallel connection of thermocouples is used for averaging.

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