Localized Cooling

Using Cold Plates

Many applications in electronics cooling require a cold plate to remove heat from discrete components laid out on a board. In these circumstances, it is more efficient that the liquid does not completely fill the cold plate, but is only transferred to areas that need to be cooled. With this kind of design, the required volumetric flow rate of the coolant will be significantly lower than if the entire cold plate was filled with liquid. The schematic for a typical example of this scenario is given in Figure 1 [1]

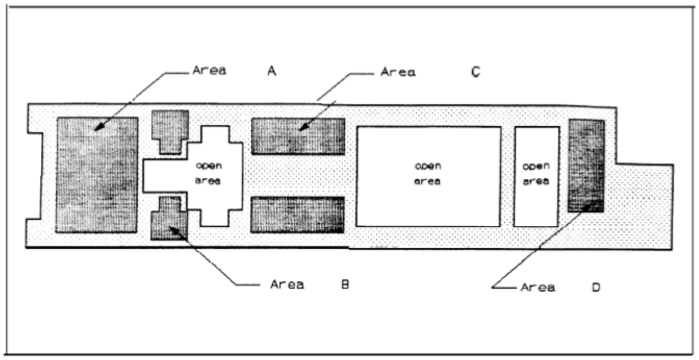

Figure 1. Schematic of a Board with a Localized Area of Heat Dissipation [1].

In Figure 1, areas A, B, C and D must be cooled for the components dissipating from 5 to 15 W/cm^2. The other areas, designated as open, have components that interfere with the cold plate and must be avoided in the design. Two designs were considered for this case: a drilled hole and a press-fit tube. Figure 2 shows the drilled hole concept. As can be seen, there are multiple small holes around the heat dissipating components under the cold plate surface. Large holes are machined to interconnect the smaller holes. A technique called gun drilling was used for machining the long holes. The entire cold plate was made from a copper block.

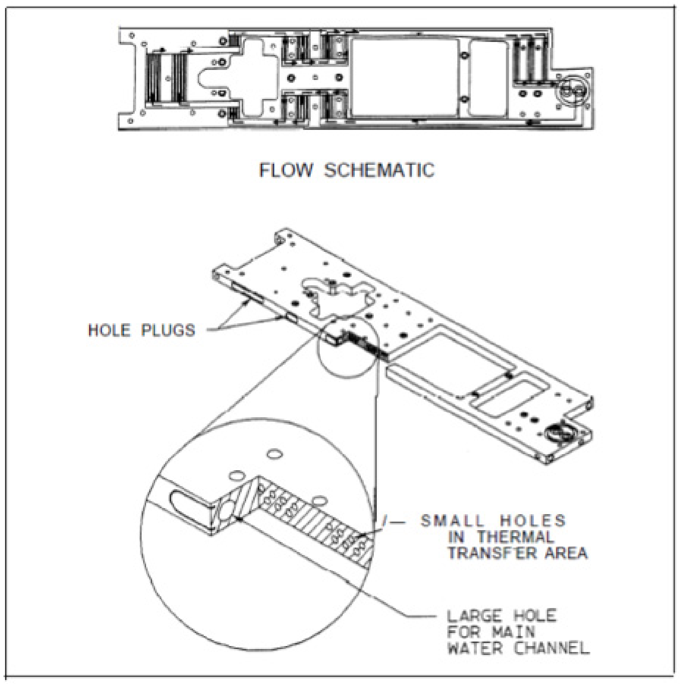

Figure 2. Schematic of the Drilled Hole Cold Plate Design [1].

Figure 3 shows the press-fit tube design. In this approach, a copper tube with high thermal conductivity is routed through the areas of heat transfer and either brazed or epoxied to the aluminum cold plate base. This design is considerably lighter and cheaper than the drilled hole design.

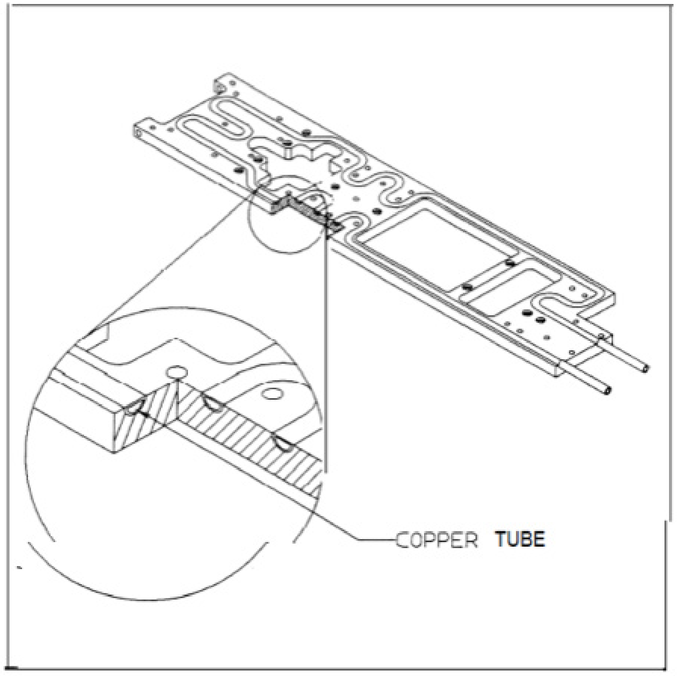

Figure 3. Schematic of the Press-Fit Tube Cold Plate Design [1].

To analyze the performance of this cold plate configuration, simple analytical tools can be used for a standard cold plate design. A brief summary of the equations is described here. To analyze the problem, we first have to calculate how much flow is going through the cold plate, and evaluate the pressure drop of the flowing fluid.

Pressure drop is calculated from:

$$\frac{\Delta P}{\rho U_m{}^2/2} = f\frac{A_{wet}}{A_c} + \sum K$$

Where
U_m = bulk mean fluid velocity (m/s)
f = fanning friction factor
A_{wet} = wetted surface area of the tube
A_c = cross section of the tube
K = loss coefficients related to turns, sudden expansion and contraction, etc.

The friction factor was obtained from the following equation which is in satisfactory agreements for the laminar, turbulent and transition regimes [2]

$$\frac{2}{f} = \left[\frac{1}{\left[(8/Re)^{10} + (Re/36500)^{20}\right]^{1/2}} + [2.21\ln(Re/7)]^{10}\right]^{1/5}$$

Where

$$Re = \frac{U_m D_h}{\nu}$$

$$D_h = \frac{4A_{wet}}{P}$$

Where ν is the kinematic viscosity of the fluid (m^2/s) and P is the wetted perimeter of the tube.
For the heat transfer calculation, the Nusselt number can be calculated from standard correlations in the literature for fully developed flow. The Nusselt number is related to the heat transfer coefficient as:

$$h = \frac{K_f . Nu}{D_h}$$

Where
K_f = fluid conductivity
For thermally developing flow the following correlation can be used [3]:

$$\frac{Nu_m}{Nu_\infty} = 1 + \frac{0.68 + 3000/Re^{0.81}}{(L/D_h)^{0.9} Pr^{1/6}}$$

Where
Nu_m = mean Nusselt number
Nu_∞ = fully developed Nusselt number
L = duct length
Then the convective resistance can be calculated as:

$$R = \frac{1}{h_m A_{wet}}$$

Where

h_m = mean heat transfer coefficient

For the tube fitted design the overall thermal resistance is made of four components: convection, tube conduction resistance, epoxy conduction resistance and the cold plate. It is stated as:

$$R = R_h + R_{tube} + R_{epoxy} + R_{coldplate}$$

Where

R_h = convection resistance

R_{tube} = conduction resistance of tube walls

R_{epoxy} = conduction resistance of the epoxy

$R_{coldplate}$ = conduction resistance of the cold plate

For the drilled design the overall thermal resistance can be written as:

$$R = R_h + R_{coldplate}$$

If the heat transfer coefficient is based on the local fluid temperature, then a caloric resistance must be added based on the fluid mass flow rate. The effective heat transfer coefficient is then:

$$h_{eff} = \frac{\dot{m}C_p\left\{1 - \exp[1/(R\dot{m}C_p)]\right\}}{A_{wett}}$$

Where

$\dot{m}$ = mass flow rate (kg/s)

Cp = fluid heat capacitance (kJ/kg·K)

Figure 4 shows the pressure drop of the two designs as a function of water flow rate. It can be seen that with a water flow rate up to 1.89 l/min (0.5 GPM) the pressure drop between the two designs is almost the same, but at higher flow rates the drilled design's pressure drop exceeds the tube design. The sharp 90 degrees turn of the drilled holes which lead to a higher loss coefficient is the major contributor to the higher pressure drop.

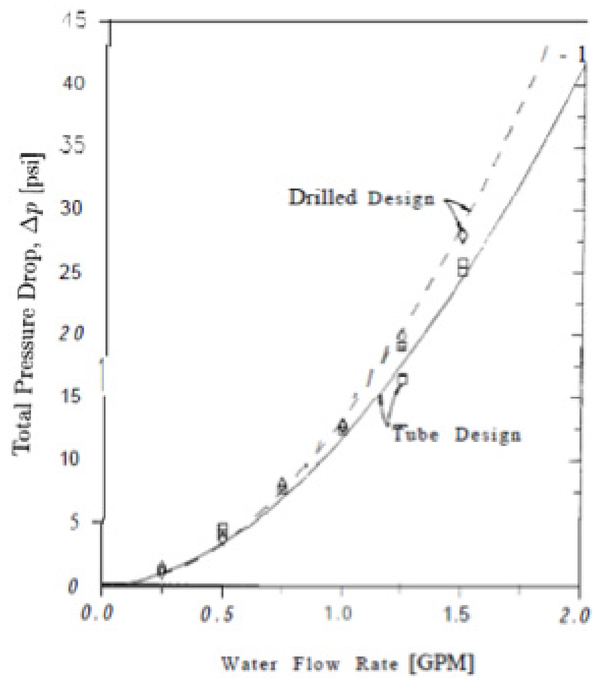

Figure 4. Total Pressure Drop of the Drilled Design and the Tube Design as a Function of the Volumetric Flow Rate [1].

Figures 5 and 6 show the effective heat transfer coefficient of the two designs as a function of flow rate. The bend and sharp increase of the curves around 0.95 l/min (0.25 GPM) is due to the flow transitioning from laminar to turbulent. The drilled hole design shows effective convection heat transfer between 7000 and 27000 W/m^2K for the range of flow between 0 and 7.56 l/m (2.6 GPM). The press-fit tube design on the other hand shows a lower effective heat transfer coefficient of between 6000 and 17000 W/m^2K. This is mostly due to the interfacial resistance and tube wall conduction. In the drilled design example, these two resistances do not exist. In a real application, the pumping of fluid is constrained by the pump and its characteristic curve. Even though the drilled hole shows a higher heat transfer coefficient for the same flow rate, the extra pressure drop caused by the drilled design may have a lower flow rate hence lowering the heat transfer coefficient.

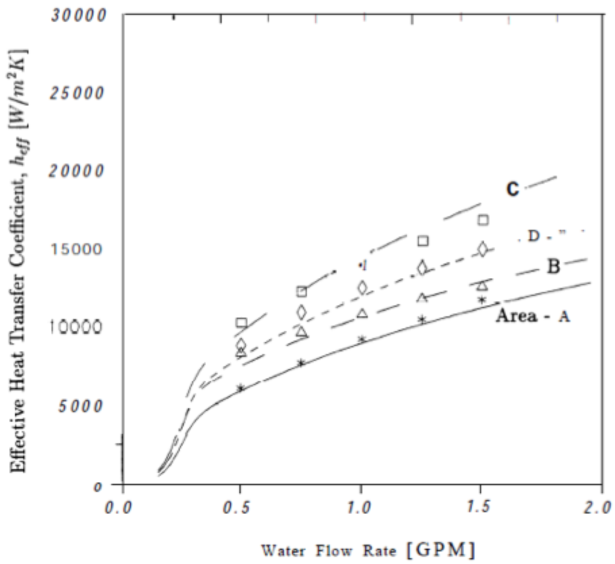

Figure 5. Effective Heat Transfer Coefficient of the Tube Design as a Function of Flow Rate for Different Regions on the Plate [1].

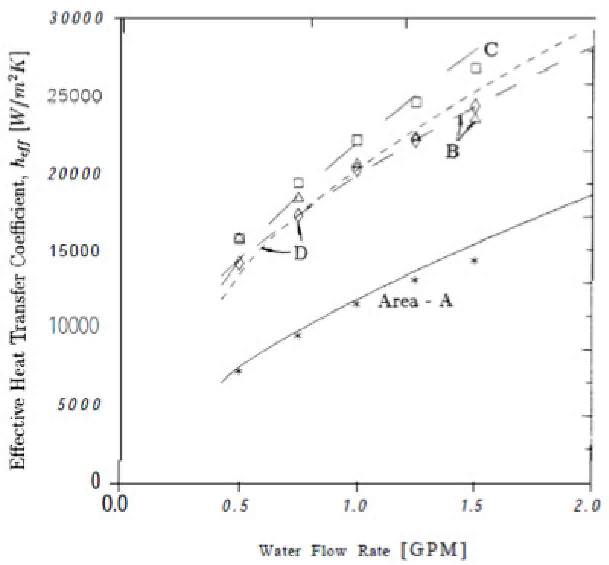

Figure 6. Effective Heat Transfer Coefficient of the Drilled Hole Design as a Function of Flow Rate for Different Regions on the Plate [1].

Figure 7 shows another cold plate design, this one by Lytron [4]. In this design, the extended-surface cold plate material and micro-channel aluminum extrusion are sandwiched between aluminum sheets. The entire assembly is welded using vacuum brazing. It is all aluminum, which makes it very light weight. The flexibility of this design allows the placement of cooling channels in different positions to enable localized cooling.

Figure 7. Lytron Vacuum Brazing of a Cold Plate for Localized Cooling [4].

The above analytics show that the performance of a cold plate for localized cooling can be calculated using a simple analytical tool. The designer then has to consider such factors as weight, manufacturing, cost and thermal performance to decide the best option for his or her design. The characteristic of the pump has a paramount effect on the design and cannot be neglected.

References:

1. Seaho, S., Moran, K. and Rearick, D. (IBM Corporation) and Lee, S. (Aavid Engineering), Thermal Performance Modeling and Measurements of Localized Water Cooled Cold Plate, http://www.aavidthermalloy.com/technical/papers/pdfs/water.pdf
2. Churchill,S., Comprehensive Correlating Equations for Heat, Massand Momentum Transfer in Fully Developed Flow in Smooth Tubes, Ind. Eng. Chem. Fundam., Vol.16, 1977.
3. Al-Arabi,M.,Turbulent Heat Transfer int he Entrance Region of aTube, Heat Transfer Eng., Vol. 3, 1982.
4. http://www.lytron.com