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Liquid Cooling for Next Generation Rugged COTS Modules

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ABSTRACT

High power/performance electronic modules are challenging the ability of air cooling to successfully remove the generated heat. Single phase liquid cooling is a proven approach for effective cooling of large amounts of heat, and has been deployed on defense platforms. Determining the thermal performance of liquid cooled cold plates can be done with basic spreadsheet calculations. These calculations can be sufficiently accurate for first order thermal analyses of design options, which enables rapid trade-off studies. To demonstrate this, a sample spreadsheet is introduced and compared to computational fluid dynamics (CFD) analyses, as well as empirical results.

INTRODUCTION

Liquid cooling of military electronics is an established approach for transferring relatively large amounts of heat in relatively small spaces, by virtue of the properties of the various preferred liquids compared to air. There are many different implementations of liquid cooling encompassing single phase cooling at the module and enclosure levels, and phase change (liquid to vapor) cooling at the module and enclosure levels. This paper will focus on single phase cooling as a starting point for obtaining the significant heat transfer benefits of liquid cooling.

Single phase liquid cooling has been implemented at both the module level and the enclosure level (and beyond, for example in base plates). At the module level, the typical implementation is known as liquid flow through or LFT, and this approach has been shown to be very effective at cooling several hundreds of Watts. For example, Curtiss-Wright and Parker-Hannifin demonstrated the ability to cool 650W total on a 0.85" pitch, 6U module, including four 150W sources representing very high power processors [1] (see Fig. 1). Today's rugged COTS modules are not at those power/heat levels yet, but the trend continues to move in that direction.

Figure 1: Liquid Flow Through (LFT) module.

Current high power/performance COTS modules are in the range of 100-200W, and standard cooling approaches employing air in either the enclosure or at the module level are at or beyond their limits. Some systems are already using liquid in enclosure sidewalls to cool rugged COTS conduction modules. Such approaches are being enhanced to cool next generation conduction modules beyond 200W.

This paper will show how these high performance modules can be cooled with single phase liquid cooling by employing straightforward spreadsheet calculators. The spreadsheet calculator has been validated against CFD (computational fluid dynamics) tools as well as test data, and shows the ease with which single phase liquid cooling can be analyzed. This contrasts with phase change liquid cooling, which is notoriously difficult when it comes to predicting cooling.

LIQUID COOLED COLD PLATE DESIGN

The liquid cooled cold plate shown in Figure 1 consists of a combination of liquid connectors, liquid manifolds, flexible microchannel coolers (for high power areas), and low power cooling areas. This highly engineered design is suitable for very high power electronic modules (e.g. 400-600W+), but engenders significant weight and cost penalties, which can be avoided for lower power modules (e.g. 200-400W).

A much simpler design, which can serve as a starting point for a low SWaP-C (size, weight, power and cost) liquid cooled cold plate, is depicted in Figure 2. This design will serve as the basis for heat transfer and pressure drop calculations that will determine its cooling effectiveness.

Figure 2: Cold Plate with parallel channels (dimensions in inches).

SPREADSHEET CALCULATOR

A spreadsheet calculator was developed using common heat transfer equations to estimate parameters of interest. The objective of the calculator was the ability to quickly calculate these parameters for a variety of scenarios to focus in on promising liquid cooled designs. More accurate analyses could then be performed using CFD (computational fluid dynamics) tools.

Once the spreadsheet calculator was developed, there was a need to determine its accuracy. Due to the simplifying assumptions required for ease of use of the calculator, it was not expected to be as accurate as more sophisticated tools like CFD, but there was a need to determine how close it would get to not only CFD, but also empirical test results.

Heat Transfer Formulae

This section will describe the main equations used in the calculator. A table showing the full set of equations is shown in the Appendix.

A key calculation performed in the spreadsheet is that of the heat transfer coefficient, h. The following equation is used to perform the calculation [2]:

$$h = J \cdot C_P \cdot G \cdot \left(\frac{C_P \cdot \mu}{k} \right)^{-2/3} \tag{1}$$

where the variables are defined in Ref. 2, page 169.

The Colburn factor, J, is defined below for laminar flow conditions (Reynolds number < 2000) [2, p. 380].

$$J = \frac{1.6}{\left(\frac{L}{D_h} \right)^{1/3} \mathrm{Re}^{2/3}} \tag{2}$$

Pressure drop calculations are also performed to understand the contribution from the cold plate to pump requirements. Equation 3 is used for this purpose [2].

$$H_L = 4 \cdot f \cdot \left(\frac{L}{D_h} \right) \cdot \left(\frac{V^2}{2g} \right) \tag{3}$$

where the variables are defined in Ref. 2, page 213.

The Fanning friction factor, f, is defined below for laminar flow conditions [2, p. 381].

$$f = \frac{16}{\mathrm{Re}} \tag{4}$$

Assumptions

Simplifying assumptions for the calculator are as follows:

- Uniform and constant wall heat flux
- Fully developed laminar flow in channels
- Negligible thermal resistance in cold plate base/fins
- Smooth walls in flow channels

Inputs

The inputs for the calculator are as follows:

- Cold plate channel dimensions – height, width, length
- Number of channels
- Fin thickness
- Power/heat input
- Liquid inlet temperature
- Liquid volume flow rate
- Liquid properties – density, specific heat, conductivity, dynamic viscosity

Outputs

The calculator provides several outputs. Below are most of the intermediate and final results:

- Mass flow rate
- Hydraulic diameter
- Weight velocity
- Reynolds number
- Colburn factor
- Convection coefficient
- Liquid outlet temperature
- Maximum wall temperature
- Heat transfer effectiveness
- Fanning friction factor
- Pressure drop

COMPARISON OF SPREADSHEET RESULTS TO EMPIRICAL TESTING

The ultimate goal of a thermal analysis is to accurately predict real-world results, therefore the spreadsheet was used to calculate results from inputs of liquid cooling tests, and then compared to the empirical results of those same tests. One such test was published by Georgia Institute of Technology [3].

The Georgia Tech work included fabrication and testing of a microchannel cooler shown in Figure 3. Sample test results are shown in Figure 4.

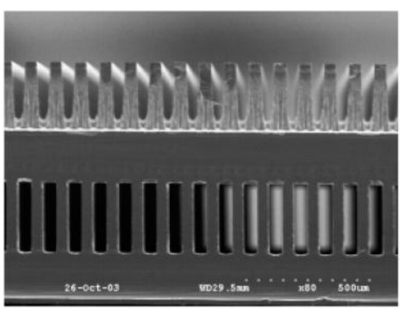

Figure 3: Georgia Tech Microchannel cooler [3].

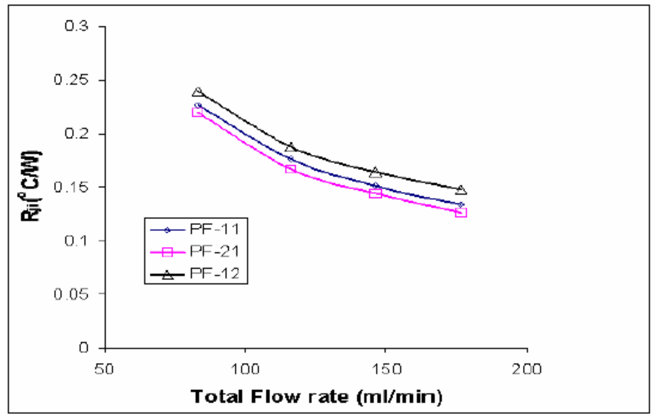

Figure 4: Georgia Tech Microchannel cooler test results.

The details of the GA Tech microchannel cooler are included in Ref. 3 and were used as input into the spreadsheet calculator. Thermal resistances of the cooler were calculated for the four test flow rates shown in Figure 4, and then compared to the test results. The comparison is shown in Table 1 below.

Flow rate (gal/min.)	GA Tech results (°C/W)	Spreadsheet results (°C/W)
0.022	0.226	0.229
0.031	0.176	0.173
0.039	0.151	0.144
0.047	0.134	0.124

Table 1: Comparison of spreadsheet results to test results.

Table 1 shows that the spreadsheet compares very favorably with the test results, with a maximum of 7.5% difference between the two (at the highest flow rate). This validation shows that the cooler in this testing behaved in agreement with the analytical prediction in the spreadsheet, providing confidence in its use.

COMPARISON OF SPREADSHEET RESULTS TO CFD

Computational Fluid Dynamics (CFD) tools are widely used to analyze thermal and fluid flow problems in order to predict useful design parameters like temperatures and pressures. As such, CFD is very useful but it requires specialized knowledge in tool use, higher performance/cost computing resources, and significant time to set up models and analyze problems of interest. The subject spreadsheet calculator was created to provide a first order analysis that avoided these drawbacks.

Accuracy of the spreadsheet relative to CFD analysis was determined by comparing results for one of the four individual coolers in a liquid cooled cold plate similar to Fig. 1. The design details of the cooler are proprietary, however other inputs for both the CFD model and the spreadsheet are as follows:

- Power/heat of 150W
- Polyethylene glycol/water (60%/40%) coolant
- 40ºC inlet temperature
- 0.07 gallons/minute flow rate

CFD results are depicted in Figure 5, with a maximum wall temperature of 63ºC.

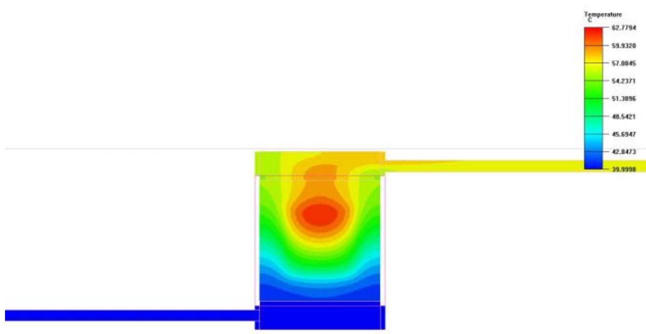

Figure 5: LFT cooler CFD temperature results

The maximum wall temperature from the spreadsheet calculation was 60ºC, which is 3ºC lower than the CFD prediction. This is in line with the results from the previous section, where higher flow rates resulted in higher discrepancies. Nevertheless, the prediction accuracy of the spreadsheet was still considered good.

The pressure drop (ΔP) was also determined using spreadsheet calculations and then compared to CFD results (Fig. 6). The CFD tool predicted a pressure drop across the cooler of 5.2 psi. The preliminary spreadsheet calculation produced a pressure drop of 4.4 psi, however it was known that the spreadsheet assumed hydrodynamically fully developed flow, and ignored entrance effects. The entrance effects were calculated using Ref. 2 as 2.2 psi, which when added to the preliminary result, gave a total ΔP of 6.6 psi. This is 27% higher than the CFD results, a significantly poorer accuracy than the temperature results. One likely cause of this discrepancy is the spreadsheet assumption of fixed coolant properties relative to inlet temperature. In reality, the coolant will increase in temperature as it flows across the cooler and absorbs heat. The higher temperature reduces coolant density and viscosity, thus lowering pressure drop.

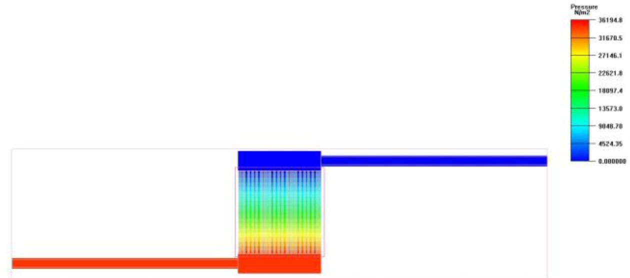

Figure 6: LFT cooler CFD pressure drop results

The next comparison was between CFD and spreadsheet results for the cold plate design shown in Figure 2. Besides the dimensions shown in Fig. 2, other pertinent inputs are as follows:

- Power/heat of 300W
  - CFD: modeled as 2 processors x 105W plus 90W additional heat
  - Spreadsheet: uniform power over the cold plate area (one side)
- PGW (60/40) coolant
- 55ºC inlet temperature
- 0.094 gallons/minute flow rate
- Overall cold plate size of 5” wide x 9.2” long

The CFD results (Fig. 7) predicted a maximum wall temperature of 65ºC, whereas the spreadsheet calculated 67ºC. This was unexpectedly close given the substantial difference in input power/heat densities, and it was also surprising that the spreadsheet overpredicted the temperature. Explanations were found by considering the heat inputs of the CFD model, details of which are proprietary.

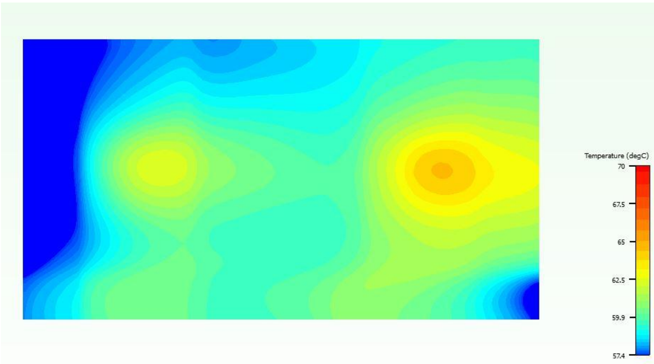

Figure 7: Cold plate CFD temperature results

The CFD analysis was also set up to calculate processor die temperatures, and the two reported values are 94.7 ºC (upstream) and 96.4 ºC (downstream). Both of these are below typical maximum allowable die temperatures of 100 ºC, providing evidence that the liquid cooled cold plate is capable of cooling high power electronic modules, in this case 300W (consisting of 2 x 105W processors plus 90W additional heat).

CONCLUSIONS

The various comparisons between the spreadsheet calculator and empirical results, and between the calculator and CFD results, clearly show the accuracy with which first order liquid cooled predictions, using basic spreadsheet calculators, can be made. These predictions allow rapid, early trade-off analyses to be performed for liquid cooled designs.

The analysis results for the simple liquid cooled cold plate design show that cooling of 200-400W electronic modules is readily achievable. This simple design can form the foundation for liquid cooled systems employing the latest in high performance electronic modules. These liquid cooled systems will be necessary to replace air cooled predecessors that are at or near their capabilities.

REFERENCES

[1] M. Benjamin and I. Straznicky, "Liquid Flow Through Coldplate Passes Muster", COTS Journal, Aug. 2006.

[2] D.S. Steinberg, "Cooling Techniques for Electronic Equipment", 2nd edition, Wiley-Interscience, 1991.

[3] Y. Joshi, "Micro-Fabricated Thermal Management Systems for Liquid Cooling of High Power Chips", 2004.

[4] W.M. Kays and A.L. London, "Compact Heat Exchangers", 3rd edition, McGraw-Hill, 1984.

APPENDIX – Spreadsheet Calculator

Description	Symbol	Formula	Units
Mass Flow Rate	M	M = VFR * ρ	lb/min
MFR per Duct	Md	Md = M / n	lb/min
Wetted Perimeter	Per	Per = 2 * (Hf + W) / 12	ft
Cross Sectional Area of airflow	CSA	CSA = W * Hf / 144	ft^2
Hydraulic Diameter	Dh	Dh = 4 * CSA / Per	ft
Weight Velocity	G	G = Md / CSA	lb/ft^2 min
Reynolds Number	Re	Re = G * Dh / μ	
Colburn Factor	J	J = 1.6/((L/Dh)$^{1/3}$ * (Re)$^{2/3}$) for laminar flow (Re< 2000)	
Convection Coefficient	h	h = (J * Cp * (G/60)) * (Cp * (μ/60) / K)$^{-2/3}$	W/ft^2 C
Total Heat Transfer Area	A	A = (2 * Hf + W) * L / 144	ft^2
Delta T from Wall to Bulk	Twb	Twb = (Q / n) / (h * A)	C
Bulk Temperature	Tb	Tb = (Ti + Te) / 2	C
Delta T from Inlet to Outlet	Tio	Tio = (Q / n) / ((Md / 60) * Cp)	C
Outlet Temperature	To	To = Ti + Tio	C
Maximum Wall Temperature	Tw	**Tw = Tb + Twb**	**C**
Effectiveness	e	e = (To - Ti) / (Tw - Ti)	
Variables			
Inlet Temperature	Ti	**input**	C
Power Dissipation	Q	**input**	W
Fin Height	Hf	**input**	in
Opening between Fins	W	**input**	in
Number of ducts	n	**input**	
Fin Length	L	**input**	in
Fin Thickness	t	**input**	in
Total Fin Core Width	Wt	Wt = (n * W) + (n * t)	in
Dynamic Viscosity of coolant at Temp	μ	**input**	lb/ft min
Conductivity of coolant at Temp	K	**input**	W/ft C
Specific Heat of coolant at Temp	Cp	**input**	J/lb C
Volume Flow Rate	VFR	**input**	CFM
Density at Temp	ρ	**input**	lb/ft^3