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THERMAL DESIGN AND PERFORMANCE OF TWO-PHASE MESO-SCALE HEAT EXCHANGERS

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ABSTRACT

Dramatically increased power dissipation in electronic and electro-optic devices has prompted the development of advanced thermal management approaches to replace conventional air cooling using extended surfaces. One such approach is Pumped Liquid Multiphase Cooling (PLMC), in which a refrigerant is evaporated in a cold plate in contact with the devices to be cooled. Heat is then rejected in an air or water-cooled condenser and the working fluid is returned to the cold plate.

Reliable, highly efficient, small-scale components are required for the commercial application of this technology. This paper presents experimental results for two-phase meso-scale* heat exchangers (cold plates) for use in electronics cooling. The configurations studied include single and multi-pass designs using R134a as the working fluid. With relatively low flow rates, low effective thermal resistances were achieved at power levels as high as 376 W. The results confirm the efficacy of PLMC technology for cooling the most powerful integrated circuits planned for the next decade.

INTRODUCTION

Integrated circuit power dissipation levels for state-of-the-art microprocessors and video processors are currently exceeding 100W with chip-level heat fluxes on the order of 50 W/cm^2. A commonly held belief by some in the industry is that further increases cannot practically be handled, and that cooling is a barrier to the ongoing progress represented by Moore's Law. We do not believe that this is true.

We have developed a thermal control technology platform that can cool chips with power dissipation of 400 W or more and corresponding heat fluxes of more than 100 W/cm^2. This technology approach, Pumped Liquid Multiphase Cooling (PLMC), can be implemented in a smaller footprint than traditional fan-cooled approaches with a cost per watt cooled equivalent to these lower-performance alternatives. It can also be used in a variety of other applications [2].

An ideal thermal management technology provides low effective thermal resistances along with a capability to move the heat some distance away from the device being cooled at high efficiency – that is, moving the heat with low parasitic power. PLMC is capable of moving a kilowatt of heat tens or even hundreds of meters away for an expenditure of a few watts of pumping power.

Figure 1 below is a block diagram of the simplest PLMC approach. Heat Q is transferred from the device being cooled via a "cold plate" or multiphase heat exchanger. While any of a number of working fluids may be used, we have chosen R134a as an appropriate design center because it is low cost, readily available, non-toxic, non-conductive, and has attractive thermophysical properties.

* In accordance with developing usage [1], we classify meso-scale channels as being in the range $3.0\ mm > D_h > 200\ \mu m$.

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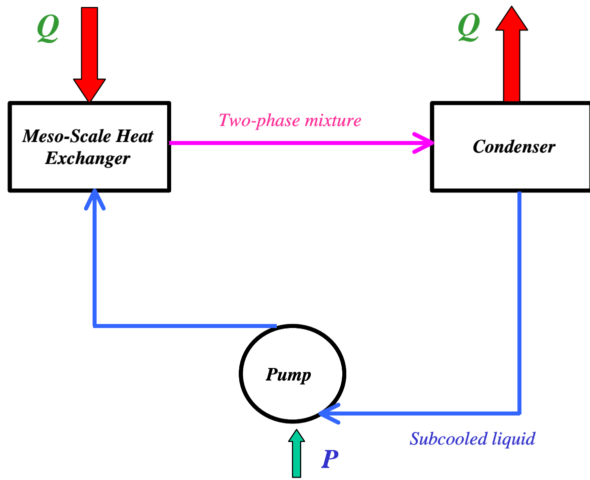

Figure 1: PLMC system

The refrigerant exits the cold plate as a two-phase mixture, typically with exit quality x of 20 – 80%. It flows to a condenser, which can be implemented in a variety of forms – natural convection cooled for a "silent" PC, air cooled, or water cooled. The slightly subcooled liquid is then pumped back to the cold plate to begin the cycle anew with the expenditure of pumping power P.

In order to be practical in typical self-contained applications (such as workstations or box/blade servers), each of the three main components must be miniaturized, high performance, reliable, and low cost. The focus of this paper is the design and performance of a meso-scale heat exchanger that can provide the requisite cooling performance in a PLMC system.

Electronics cooling devices are typically characterized using the concept of thermal resistance – that is, the temperature rise across a component for a given heat transfer, in, for example, kelvins per watt or ^{o}C/W. In simple terms, the overall thermal resistance for a PLMC system can be written as

$$R_{tot} = R_{ds} + R_{sv} + R_{va} \,. \tag{1}$$

Here the total resistance is made up of a resistance from the integrated circuit die to the heat sink (cold plate) R_{ds} plus the resistance in the micro-scale heat exchanger R_{sv} (sink to saturated vapor) plus the condenser resistance R_{va}. In most cases R_{ds} is constrained by packaging and interconnection requirements, while the resistances associated with the PLMC system must be sufficiently low as to ensure an operating temperature below, say, 100 ^{o}C*.

* Dependent on manufacturer and application.

For today's chip packaging approaches R_{ds} is of the order of 0.15 K/W and ambient temperatures can be 40 ^{o}C or higher. Assuming a 200W processor, a combined heat exchanger and condenser resistance of 0.15 K/W is therefore required. Here we assume that half of this resistance, or 0.075 K/W, can be allocated to the cold plate.

A series of two-phase cold plate designs have been built and tested by Thermal Form and Function that can meet these stringent requirements. In this paper, we present experimental results for the thermal and pressure-drop performance of three of these designs.

The details of the flow boiling heat transfer within these devices are exceedingly complex, and correlations and design data are not available in the literature. In particular, the two-phase flow dynamics are not easily characterized, and the orientation, heat source size, heat flux, and flowrate are important variables.

The results presented in this paper should be viewed as preliminary results of a comprehensive study focused on the engineering design optimization of these high performance meso-scale cold plates.

MESO-SCALE HEAT EXCHANGER DESIGN

There has been considerable investigation in recent years of 100 µm scale "microchannels" that can provide high thermal performance for miniaturized heat exchangers for electronics cooling [1,3]. While good thermal performance can be achieved, there are several drawbacks to the practical implementation of this technology. First, there is at present no convenient mass production process for low cost microchannel implementations – while it is intriguing to consider silicon heat exchangers, for example, the fabrication costs are not insignificant and the capital costs are substantial.

In order to achieve high performance with a single-phase coolant, high heat capacity and conductivity are needed; thus water is the fluid most often used. This limits the choice of materials in the cooling system due to corrosion potential and results in concerns about fouling. Finally, to take advantage of the significant benefits of multiphase cooling, serious problems with flow instabilities in the channels must be dealt with.

The two-phase heat exchanger designs we have developed are by contrast easy and cost effective to manufacture. They consist of a high thermal conductivity shell with internal fins and can be single- or multi-pass. We have observed no flow instabilities, no doubt due to the larger geometries employed as well as the more benign thermodynamic properties of R134a.

A two-pass design of the type used for the experimental results presented herein is shown in Figure 2. The materials used can be varied, and mixed in a particular design, since there are no corrosion issues. For the work reported here, we have used an all-copper construction for both the shell and the internal fins. The device consists of three basic parts: a base with interior flow passages (along with headers), an easily fabricated fin structure, and a top cover plate. The assembly (including inlet and exit tubing) is brazed. The fin structure is brazed to both the cover plate and the base.

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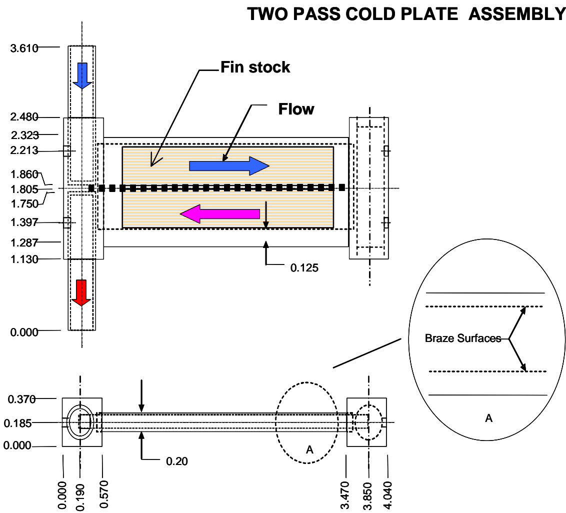

Figure 2: Micro-scale heat exchanger (dimensions in inches)

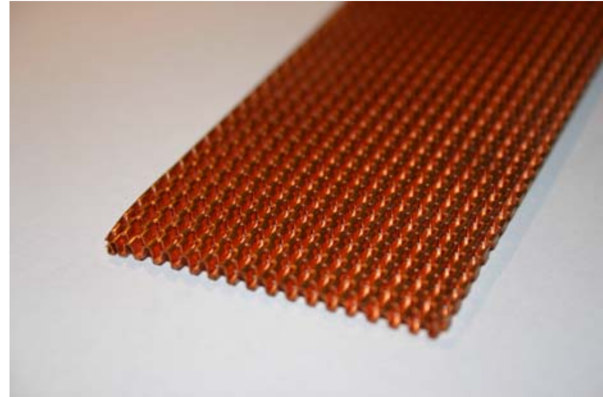

Figure 4: Offset strip fin

The fin geometry for the results presented here is shown in Figure 3. The internal fin structure is periodically offset to facilitate mixing and cross-exchanger flow distribution. A photograph of the fin stock used is shown as Figure 4.

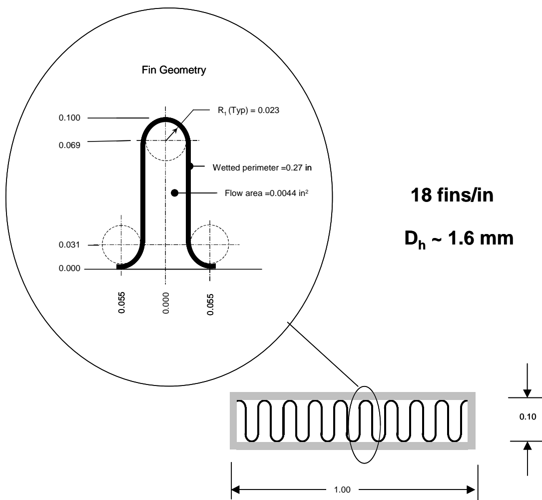

Figure 3: Fin geometry (dimensions in inches)

EXPERIMENTAL APPARATUS

In most product implementations, PLMC systems will operate in harmony with the environmental conditions – that is, the system pressure and saturation temperature will change as ambient conditions change. In order to characterize heat exchanger designs, condenser designs, and micro-pumps, we developed an experimental rig that maintains constant system conditions while measurements are being made. The experimental setup is shown in Figure 5 below.

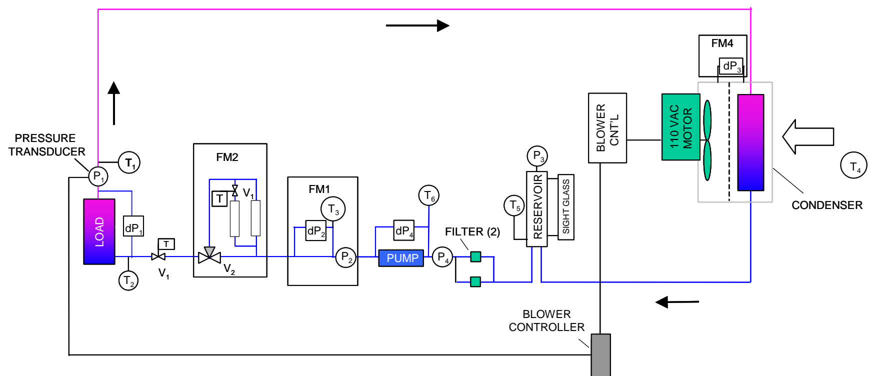

Figure 5: Experimental rig

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To provide an accurate simulation of the power load from a typical packaged microprocessor, the heater / coldplate stackup shown in Figure 6 was used. Heat is generated in a foil heater element sandwiched between copper contact plates and phenolic thermal insulating material. Indium gallium is used to provide a low-resistance heat flow path from the heater elements to the micro-scale heat exchanger. The actual area of contact is controlled using a machined pedestal on the copper heater block; for the experiments reported here, the area was 2 cm X 2 cm. The base of the micro heat sink in these tests was 0.070 in. thick. Five thermocouples were mounted in laterally drilled holes in the base with centerline 0.035 in. from the heat sink surface.

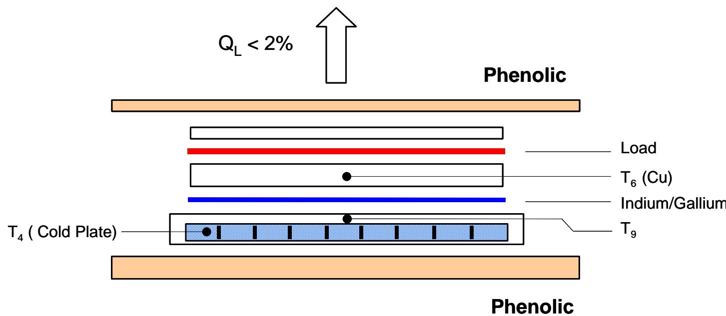

Figure 6: Heater block layup

Using this experimental apparatus, differential pressures were measured with an accuracy of 0.0125 psi (0.007 MPa) and temperatures were measured with sheathed thermocouples with a repeatability of +/- 0.1 ^{o}C. A careful analysis and confirming experiment showed that heat loss from the top and sides of the stack was less than 2% of the total heat load, the remainder being delivered to the working fluid in the cold plate. (This was accomplished by measuring the power required to maintain the cold plate at the anticipated operating temperature with no flow.) A custom flow meter was used with an estimated accuracy of +/- 5%.

EXPERIMENTAL RESULTS

Cold plates with 1, 2, and 3 passes have been fabricated and tested, and thermal resistances have been measured with heated areas of 1 cm X 1 cm and 2 cm X 2 cm. Heat fluxes of up to 376 W/cm^2 have been employed.

In this paper we report results for cold plates with the external dimensions of Figure 2 with a 2 cm X 2 cm heated area and 1, 2, or 3 coolant passes. In all experiments, heat was imposed on the top of the cold plate. We have confirmed that there is a slight effect of orientation on the heat transfer results and are continuing work to quantify these effects. Identical fin structures were used in each case, with a passage hydraulic diameter of ~ 1.6mm*.

The thermal resistance data we present below represents the "sink to saturated vapor" resistance R_{sv}. The "sink temperature" we have employed is the maximum recorded thermocouple reading in the cold plate, occurring at the center of the heated area, and the saturation temperature of the working fluid at the cold plate exit.

As defined and measured, R_{sv} includes both a conduction resistance term due to the 0.035 in. thick copper layer between the thermocouple location and the interior wall of the cold plate and the flow-boiling related thermal resistance from the wall to the working fluid. That conduction resistance is of the order of 0.01 K/W. Since the purpose of this work was to derive thermal design data for PLMC systems based on the simple resistance model of Equation 1, we have chosen not to confuse the issue by reporting results of wall-fluid resistance based on wall superheat. In future publications focused on the actual flow-boiling phenomena, we will do so.

In Figure 7, thermal performance data for the 1, 2 and 3 pass cold plates is shown as a function of flow rate at a heat flux of 94 W/cm^2. Flow rates for PLMC systems are comparatively low; our designs focus on flowrates between 1 and 4 gallons/hour (63 to 252 ml/min).

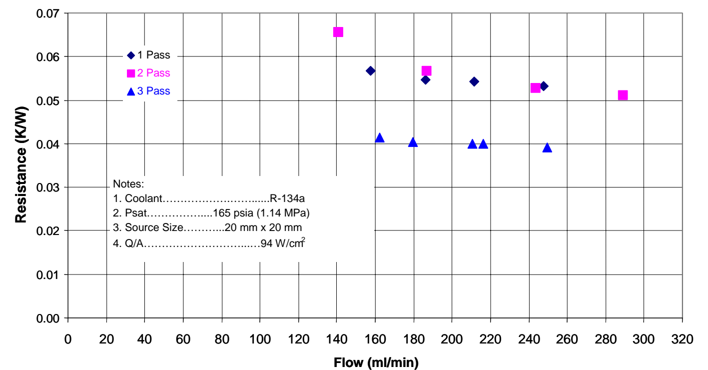

Figure 7: R_{sv} vs. flowrate for 1, 2, and 3 pass cold plates

We note that the performance of the 1 and 2 pass designs is similar, but the 3-pass design provides a performance improvement of about 20%. We attribute this to fluid velocity effects; the 3-pass design has lower overall cross sectional area for flow.

Figure 8 shows the relationship between thermal performance and heat flux for the 3-pass design. There is a relatively strong dependence of R_{sv} on heat flux, presumably the result of the effects of convective boiling. The thermal resistances measured meet the design target of 0.075 K/W; for a 200 W load (50 W/cm2), the sink-to-fluid temperature difference is of the order of 11 ^{o}C at the lowest flowrate.

* Note, however, that the fins were of the offset variety, promoting mixing and allowing fluid communication between channels.

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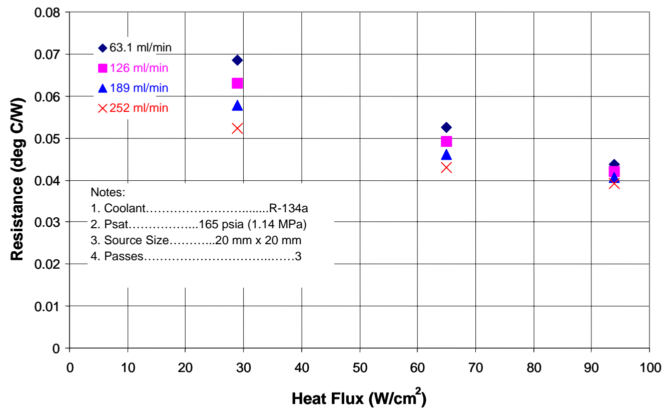

Figure 8: R_{sv} vs. heat flux for the 3-pass design

Figure 9 presents essentially the same thermal performance results as in Figure 7 plotted as a function of exit quality (flowrate varied; heat load of 376W). The dependence of thermal performance on exit quality is slight for this set of thermal and flow conditions – overall, a useful characteristic for electronics cooling applications.

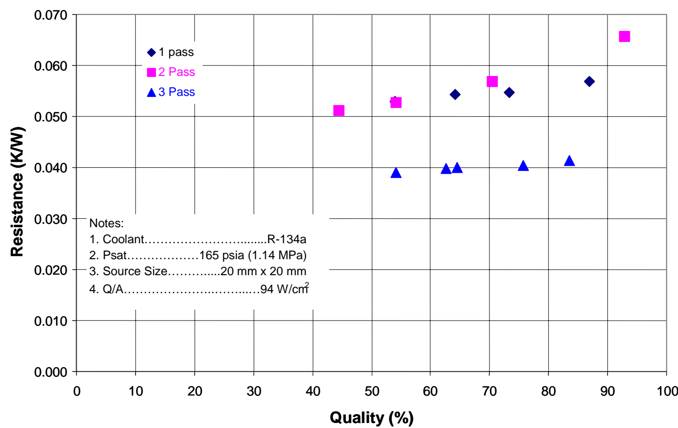

Figure 9: R_{sv} vs. exit quality

Finally, Figure 10 presents pressure drop data for the 1, 2, and 3 pass designs, again with a heat load of 376 W. The trend is as expected, with a roughly square-law dependency on velocity despite varying void fractions. Of real interest and import are the very low levels of pressure drop, of the order of a few psi. This means that a small, very low power pump can be used for a high performance PLMC system. "Coefficients of performance" – that is, the ratio of heat transferred to pumping power, are in fact of the order of 100 for systems on this scale.

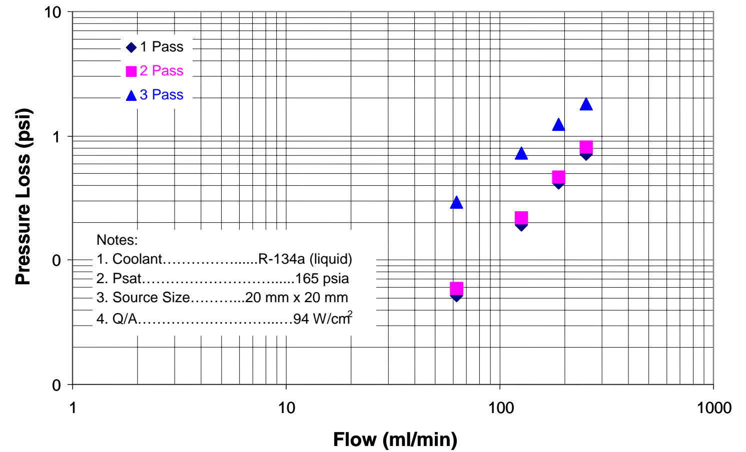

Figure 10: Pressure drop data for multipass cold plates

CONCLUSION

The thermal performance required to cool microprocessors and other electronics to power levels of at least 376W has been demonstrated using R134a in a meso-scale flow boiling heat exchanger appropriate for use in a practical PLMC thermal management system. The results are similar to those reported for microchannel cooling using water as the coolant [3], with a pressure drop (and thus pumping power) an order of magnitude smaller. No flow instabilities were observed with exit qualities in excess of 80%. Manufacturing for these devices is straightforward.

The two-phase cold plates described do not necessarily represent optimized designs. In particular, varying the internal fin design and optimizing flow parameters for various heat source sizes may result in significant improvements over the already acceptable results presented here. Optimization work, flow visualization, and development of an overall framework theory are underway and will be presented in future papers.

The designs and concepts described in this paper are protected by issued and pending US patents.

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NOMENCLATURE

P	Pump power (W)
Q	Thermal load (W)
Q_L	Heat lost from experimental heater (W)
R_{tot}	Die to ambient thermal resistance (K/W)
R_{ds}	Die to heat sink resistance (K/W)
R_{sv}	Sink to saturated vapor resistance (K/W)
R_{va}	Condenser resistance (K/W)
x	Vapor quality exiting cold plate (%)

REFERENCES

[1] Steinke, M. and S. Kandlikar, "an Experimental Investigation of Flow Boiling Characteristics of Water in Parallel Microchannels," *Journal of Heat Transfer* **126**, 518 – 526, August 2004.

[2] Hannemann, R., Marsala, J. and Pitasi, M., "Pumped Liquid Multiphase Cooling," ASME paper IMECE2004-60669, November 2004.

[3] Pokharna, H. et al, "Microchannel Cooling in Computing Platforms: Performance Needs and Challenges in Implementation," ASME Conference on Microchannels and Minichannels, June 2004.

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