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Charles Coggins

Georgia Institute of Technology

David Gerlach

Georgia Institute of Technology

Yogendra Joshi

Georgia Institute of Technology

Andrei Fedorov

Georgia Institute of Technology

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Compact, Low Temperature Refrigeration of Microprocessors

Charles COGGINS^{1*}, David GERLACH², Yogendra JOSHI³, Andrei FEDOROV⁴

Georgia Institute of Technology,
The George W. Woodruff School of Mechanical Engineering,
Atlanta, Georgia, USA

¹Phone: (404) 993-2506, Fax: (404) 894-8496, E-mail: gth629z@mail.gatech.edu

²Phone: (404) 385-1878, Fax: (404) 894-8496, E-mail: david.gerlach@me.gatech.edu

³Phone: (404) 385-2810, Fax: (404) 894-8496, E-mail: yogendra.joshi@me.gatech.edu

⁴Phone: (404) 385-1356, Fax: (404) 894-8496, E-mail: andrei.fedorov@me.gatech.edu

ABSTRACT

A compact, two-stage R-404A / R-508b cascade refrigeration system designed for interface with a high-performance microprocessor is under development. Such a system, designed to remove 100 W at -70 °C, can avoid the high cost of deep cryogenic cooling. Because thermal interface design is critical in maintaining low chip temperatures, a compact, microchannel evaporator is used. A chemically microetched copper interstage heat exchanger that uses branching flow paths for improved refrigerant distribution is under development. The initial temperature lifts, capacity, and physical size of the system are reported. Future work to refine the evaporator design and to further miniaturize the system through compressor design is planned. The ultimate goal is to build an inexpensive, quiet, reliable system, small enough for integration with a desktop computer, that can obtain a chip operating temperature of -70 °C.

1. INTRODUCTION

Microprocessor performance can be improved by lowering the junction temperature. Naeemi and Meindl (2004) show that, due to a decrease in leakage current at low temperature operation, a CMOS chip has the potential of achieving a 4.3X performance enhancement at -100 °C, compared to 85 °C operation. In addition to enhanced chip throughput, sub-ambient cooling offers orders of magnitude improvement in reliability (Schmidt and Notohardjono, 2002). While miniature cryogenic systems are capable of very low temperatures, they are exceptionally expensive to purchase and operate. Two-stage cascade vapor compression refrigeration (VCR) is a compact, reliable, and simple technology that can achieve temperatures close to -100 °C at a cost that is orders of magnitude below existing cryogenic systems. For these reasons, the authors are developing a miniature cascade VCR system designed to interface with a high performance microprocessor.

Over the past decade, the performance of laptop, desktop, and server computers has undergone vast improvements due to miniaturization of CMOS technology and faster clock rates. However, in order to meet this functionality, heat dissipation rates have rapidly increased (SIA, 2005). It will become difficult to meet these thermal management demands with either conventional air-cooling or non-chilled liquid cooling. Refrigeration may provide the only means by which future high performance processors can be maintained below predicted maximum temperature limits (Phelan *et al.*, 2001). With over 100 years of reliable operation in various applications, VCR is an obvious choice to consider for electronics cooling (Schmidt and Notohardjono, 2002). Because it can remove large heat loads at below ambient temperatures, VCR systems are already being used to cool computer and telecommunications equipment in some high performance applications (Peeples, 2001). However, until VCR systems can be made compact enough to easily integrate into packaging, widespread application of refrigeration in electronic cooling will remain limited (Phelan *et al.*, 2001).

A cascade VCR system that can offer temperatures as low as -100 °C could be an inexpensive, compact alternative to cryogenic technologies such as Gifford-McMahon and Pulse Tube cycles (Wadell, 2005), and would avoid the cost of expending cryogenic fluids in open cooling cycles (Schmidt and Notohardjono, 2002). Furthermore, temperatures of 173K-233K present less significant electronic packaging problems to overcome than does operation at deep cryogenic temperatures (Peeples, 2001). Because of these advantages, VCR systems could make low-temperature operation a commercial reality for medium-size computers and large servers.

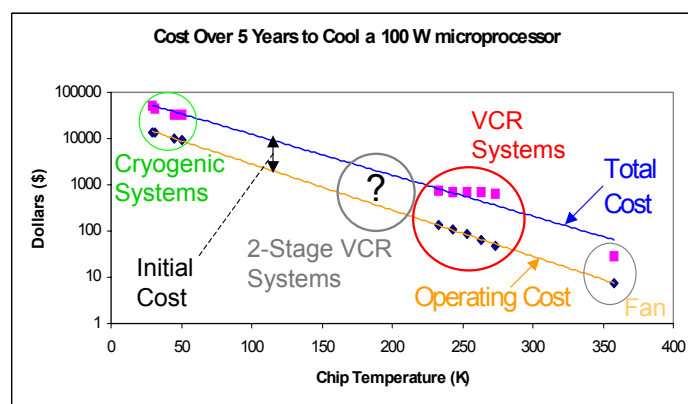


Figure 1: Cost over 5 years to cool a 100 W microprocessor (Wadell, 2005)

In recent years, both industry and academia have approached the design, modeling, and construction of small VCR systems for microprocessors. The first high-end system to use refrigeration was the IBM S/390 G4 CMOS system, first shipped in 1997. With two Modular Refrigeration Units (MRU), each fitting inside a 6 U package (1 U = 1.75 in.), the system was designed to dissipate 1050 W at 35 °C from a G4 multichip module. The COP of the system varied from 2 to 3 depending on the environmental conditions. Schmidt and Notohardjono (2002) designed cold plates to interface the VCR system with the chips and dealt with condensation through the use of desiccants and strategically located insulation. While the authors acknowledged that refrigeration is more costly and uses more power than more basic thermal management techniques, these disadvantages were offset by gains in performance and reliability. Asetek, Inc. (2006) has sold personal computer cooling solutions utilizing refrigeration for several years. A system in Asetek's VapoChill line of products can dissipate 200 W of heat at -33 °C using R-507. Researchers have explored designs to reduce the size of such systems, thus making them more attractive for electronics packaging. Chow *et al.* (1999) presented a preliminary design and analysis of a mesoscale refrigerator to be created from layers of silicon wafers bonded together and fabricated through the techniques of microelectronics. The system was designed to remove 32 W at 12 °C, with a COP of 3.34, using a centrifugal compressor driven by an electrostatically actuated, pancake-shaped motor. Heydari (2002) evaluated a CPU spot cooling refrigeration system and developed a thermodynamic model describing the performance of the cycle. The system was designed to cool an 86 °C chip junction to a 20 °C evaporating temperature using a miniature, oil-less, linear reciprocating compressor with an estimated COP of 3.0. By limiting the system to a compressor, capillary tube, compact condenser, and cold plate evaporator heat exchanger, the system volume was minimized. Phelan *et al.* (2004) explored the possibility of developing a mesoscale VCR system, less than 5 cm in size, that could be integrated into high power microelectronics packaging. While the authors determined that heat loads of up to 300 W could be dissipated at 5 °C using commercially available scroll compressors, the compressor itself could not fit inside the desired 5 cm package. These systems are summarized in Table 1.

The current work is the second phase of a three-phase project with the ultimate goal of developing a miniature refrigeration system that can remove 100 W/cm² at -100 °C. The final system must be reliable, technologically simple, and sufficiently compact to be integrated with a high performance desktop microprocessor. During the first phase of the project, a two-stage R-134a / R-508b cascade refrigeration system was developed that can remove 100 W/cm² at -62.6 °C. However, this system was used only to test evaporator designs, and at about 1 m³, the system was not optimized for size. The objective of the current phase of the project is to reduce the size of the system by an order of magnitude, while maintaining performance, using off-the-shelf parts and improved component matching. Size is the priority metric for future work. The micromilled evaporator from the first phase is retained, and a chemically microetched copper stage-to-stage heat exchanger is under development.

Table 1: VCR systems used in electronics thermal management

	Schmidt and Notohardjono (2002)	Asetek, Inc. (2006)	Chow <i>et al.</i> (1999)	Heydari (2002)	Phelan <i>et al.</i> (2004)	Wadell (2005)
Heat Load [W]	525	200	32	–	100-300	100
T_{evap} [°C]	15-35	–33	12	20	5	–62.6
T_{cond} (T_{amb}) [°C]	–	–	(45)	60	55	(25)
Flow Rate [g/min]	–	–	16.3	–	–	70
Number of Stages	1	1	1	1	1	2
System Size	267x267x711 mm, 27 kg	210x210x490 mm, 15 kg	Mesoscale	–	~150x50x50 mm	~1m ³
Refrigerant	R-134a	R-507	R-134a	R-134a	R-134a	R-134a / R-508b
COP	2-3	–	3.34	3.0	~3	–
Compressor Type	DC Rotary	AC Reciprocating	Centrifugal	Linear	Scroll	AC Reciprocating

2. SYSTEM MODEL AND REFRIGERANT SELECTION

2.1 System Model

In a cascade system, two separate refrigerant systems are interconnected, with the evaporator from the high stage used to cool the condenser of the low stage. This allows the low stage to operate at a lower temperature and pressure than possible with the same size single-stage system.

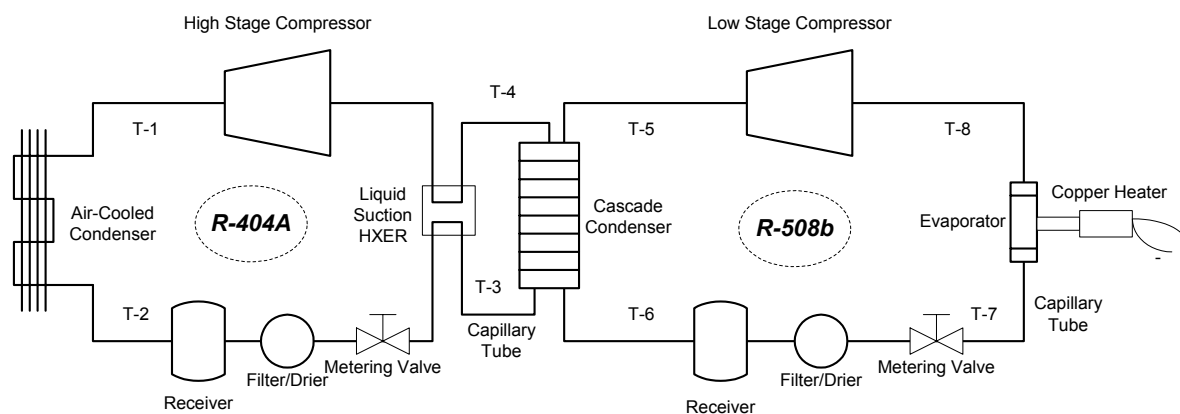


Figure 2: Cascade Refrigeration System Schematic

A simple model of a two-stage cascade VCR system was developed in Engineering Equation Solver (EES) (Klein, 2005). The air-cooled condenser, cascade condenser, and liquid-suction heat exchangers were modeled using the Effectiveness-NTU Method (Incropera and DeWitt, 2002). Figure 3 shows the T-s diagram developed for the system, and Table 2 lists the compressor requirements that were developed from the model.

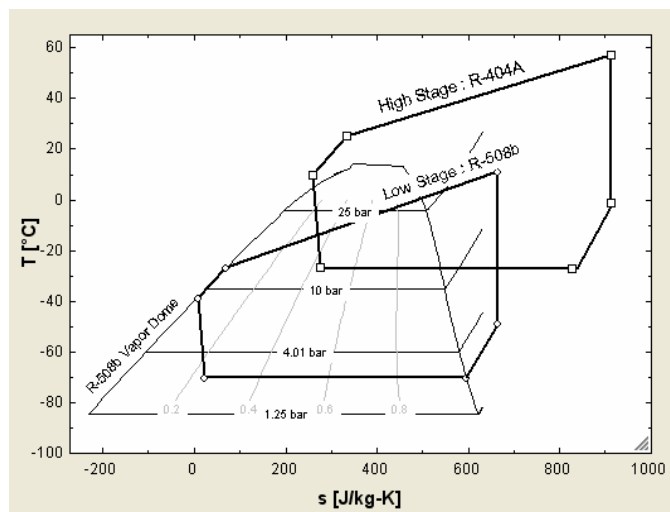


Figure 3: EES Model

Table 2: Compressor Requirements developed from EES Model

	Capacity [W]	Mass Flow Rate [g/min]	T_{evap} [°C]	$T_{\text{cond}} (T_{\text{amb}})$ [°C]	Required Pressure Ratio
High Stage (R-404A)	190	83.9	-26.4	57.4 (25)	5.5
Low Stage (R-508b)	100	51.7	-70	11.4	5.3

2.2 Refrigerant Selection

In the first phase of this project, R-134a was chosen for the high stage because it is widely available. In the current phase, R-404A was chosen over R-134a. With a lower boiling point, R-404A is more suited for low temperature applications. R-508b is commonly used as the refrigerant in the low stage of cascade systems.

3. REFRIGERATION SYSTEM AND EXPERIMENTAL SETUP

3.1 Refrigeration Flow Loop

A two-stage cascade VCR system, shown in Figure 2, is used to deliver R-508b to a compact microchannel evaporator. The evaporating R-404A in the high stage removes heat from the condensing R-508b refrigerant in the low stage. Each stage uses a brushless DC hermetic compressor, chosen for its high capacity rating and small shell size. An air-cooled condenser on the high stage and a cascade condenser on the low stage remove the condensation heat from the refrigerant vapor. A receiver placed after each stage's condenser serves as a storage container for liquid refrigerant that enters from the condenser. Following the receiver and a filter/drier, a metering needle valve and a capillary tube throttle the refrigerant in each stage. On the high stage, the capillary tube is wrapped around the suction line to create a liquid-suction heat exchanger which subcools the liquid refrigerant, boils liquid refrigerant in the suction line, and reduces flash gas in the liquid line. The compressors and components were selected to minimize the overall size of the system. The compressors, condenser, cascade condenser, and receivers fit within a 320x320x540 mm box.

Table 3: Cascade Refrigeration System Component Sizes

<i>All sizes in mm</i>	Compressor	Receiver	Fliter / Drier	Air-Cooled Condenser w/fan	Cascade Condenser	Insulated Evaporator
High Stage	100x134x209	90x90x200	20x20x100	180x200x280	40x40x340	–
Low Stage	100x134x209	90x90x200	20x20x100	–		50x90x90

3.2 Cascade Condenser

While the cascade condenser is not a significant fraction of the total system volume, it can be reduced in size more easily than the compressors or the air-cooled condenser. Although the tests described in this paper used a conventional concentric tube heat exchanger, a chemically microetched copper stage-to-stage evaporator/condenser is under development. Various analytical, computational, and experimental studies have shown that dendritic or fractal flow paths can reduce the pressure drop needed to distribute single phase fluid flow in heat exchangers (Bejan, 2000). In addition, prior studies have shown milled offset pin fins are effective in evaporation (Wadell, 2005). A combination of these designs is under development. The heat exchanger is composed of a series of stacked plates chemically etched from copper. A representative photolithography mask used in fabricating the plates is shown in Figure 4. This design utilizes branching channels to minimize maldistribution without a significant pressure drop penalty. In addition, the passages interconnect within the heat exchanger to prevent the dryout of individual channels and the resultant instability. The widths of the individual channels decrease as their number increases. The minimum channel size in the current design is approximately 0.750x0.150 mm, and the total size is 68x93 mm. The design will be optimized to minimize the total cascade cycle system size. The heat exchanger performance will be measured during the operation of the cascade cycle.

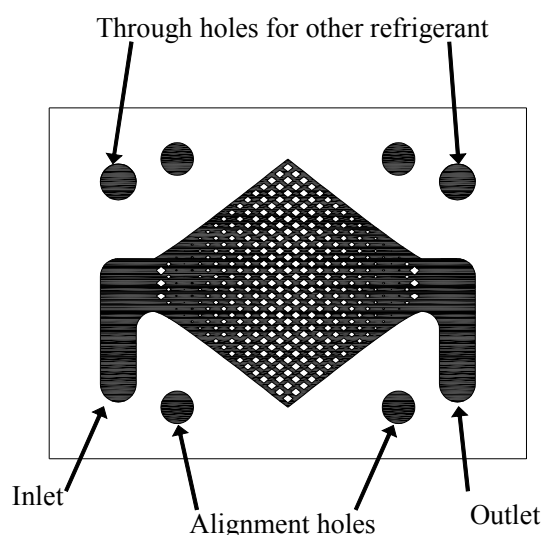


Figure 4: Schematic of interstage heat exchanger plate.

3.3 Low Stage Cold Plate Evaporator

The cascade refrigeration system must interface with the microprocessor by means of a high heat flux, low temperature, compact cold plate evaporator, appropriately insulated to avoid condensation or freezing of water on electronic components. The cold plate is fabricated from copper and has thin walls designed to shorten the thermal path from a copper heat block, which substitutes for a 1 cm² microprocessor. The heat block contains two 100 W cartridge heaters powered by a variable AC transformer. The heat block junction temperature is measured by three thermocouples inserted inside the block, located 1 mm from the cold plate interface. Three additional thermocouples, evenly spaced along the length of the heat block are used to confirm a 1-D temperature gradient to the evaporator. Thermal grease is used to reduce thermal resistance between the cold plate and the heater block, and a clamp is used to increase contact pressure. While a solder joint between the cold plate and copper heat block would greatly reduce thermal resistance, such a connection could fracture a silicon microprocessor, which has a much smaller thermal expansion coefficient than copper. The evaporator internal design optimizes heat transfer by increasing both surface area and the length of the refrigerant path. A 0.4 mm wide, 1.2 mm deep end mill was used to mill the evaporator shown in Figure 5. The microchannel evaporator contains two manifolds that are 10x10 mm and 1.2 mm deep, and a 10x10 mm area to evaporate the refrigerant. There are 13 channels, each 1.2 mm deep and 400 μ m wide, with a spacing of 800 μ m. The hydraulic diameter is 600 μ m, and the microchannel cross-sectional area is 6.24 mm² (Wadell, 2005).

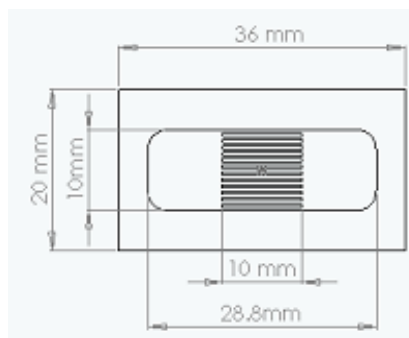


Figure 5: Top view of the microchannel evaporator (Wadell, 2005).

4. RESULTS AND CONCLUSIONS

During initial tests of the system, the low stage delivered R-508b to the evaporator at average sustained temperatures of $-76.6\text{ }^{\circ}\text{C}$, but the system lacked the desired capacity. Under a heat load of 34 W, the surface of the copper heat block was cooled to an average temperature of $-2.8\text{ }^{\circ}\text{C}$. The authors believe that lower chip temperatures could be obtained by improving the thermal interface between the cold plate and the copper heat block. Some enhancement in this interface could be achieved by reducing the contact resistance through more even compression and more uniform application of thermal grease. The thermal interface could be further improved through alternate compact evaporator designs that utilize different enhancement structures with optimized geometries. In addition, there are still some gains to be made in system efficiency and capacity. These gains could be realized by minimizing heat leaks through improved application of insulation and by desuperheating the hot gas discharge from the low stage before it enters the cascade condenser.

The physical size of the current system is about 0.1 m^3 , or one order of magnitude smaller than the system developed in the first phase of the project. Adding a microfabricated cascade condenser will reduce system size somewhat, but to ultimately achieve the goals of this project, smaller and more powerful compressors are required. This sentiment was echoed by many of the authors listed in the introduction of this paper (Phelan *et al.*, 2004; Heydari, 2002; Unger and Novotny, 2002). Both scroll and rotary compressors will be considered for use in a future system.

NOMENCLATURE

COP coefficient of performance (–)

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