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Review of Refrigeration Technologies for High Heat Dissipation Electronics Cooling

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REVIEW OF REFRIGERATION TECHNOLOGIES FOR HIGH HEAT DISSIPATION ELECTRONICS COOLING

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ABSTRACT

It is envisioned that conventional air cooling techniques using heat sinks are not able to sufficiently cool future generations of high heat dissipation semiconductor processors. Thus, alternative cooling approaches are being extensively studied as substitutes. This paper presents a literature review of the available refrigeration systems and system simulation models for electronics cooling. The results of this review are used to facilitate an ongoing effort by the authors to develop and design a Miniature-Scale Refrigeration System (MSRS) for high heat dissipation electronics cooling as well as the development of an MSRS simulation model.

1. INTRODUCTION

Thermal management of electronic components is the primary concern in developing faster, more compact and reliable computer technology. In particular, the cooling of future electronic chips is essential. Based on Moore's prediction [1], the number of transistors on integrated circuits currently doubles approximately every eighteen months. As the number of transistors in a single chip increases, the power consumption and heat dissipation from the chip will become a critical issue in the design of high performance semiconductor processors. The International Technology Roadmap for Semiconductors 2003 [2] predicted that the heat dissipation of a single chip package will be 170 W in 2005 for high performance systems. It also notes that the maximum junction temperature should be 85°C for both cost performance and high performance systems.

A conventional air cooling technology has been used for several years by using a heat sink and fins to increase the surface area to dissipate the heat from the heat source to the ambient air. This air cooling approach will no longer be sufficient as the power consumption of the CPU increases and the size decreases. Therefore, thermal management engineers have searched for alternative cooling approaches to maintain the chip temperature below the maximum operating temperature. Alternative cooling methods are, for instance, heat pipe technology, jet impingement cooling, thermoelectric cooling, liquid cooling, spray cooling, and refrigeration cooling.

A refrigeration cooling technique is one of the most promising cooling techniques to keep the junction temperature below the maximum operating temperature and handle high heat dissipation applications. The advantages of the refrigeration cooling technique [3] are: (1) to maintain a low junction temperature and at the same time dissipate high heat fluxes; (2) to increase the device speed due to a lower operating temperature; and (3) to increase the device reliability and life cycle time because of a lower and constant operating temperature. The disadvantages of the refrigeration cooling technique are: (4) an increased complexity and cost; (5) the need for additional space to fit the components of the refrigeration system; and (6) a decrease of the system reliability as a result of an additional moving component, i.e., the compressor.

Based on the advantages listed above, an effort was started by the authors to develop a miniature-scale vapor compression system for electronics cooling. As part of this effort, the currently available vapor compression systems and system simulation models for electronic cooling were reviewed. A summary of this review is given in this paper.

2. TYPES OF REFRIGERATION SYSTEM FOR ELECTRONICS COOLING

Scott [4] classified refrigerated cooling of electronic equipment into four main types: 1) refrigerated cooling of air or liquid; 2) refrigerated heat sinks; 3) liquid nitrogen baths; and 4) thermoelectric coolers. He suggested that the refrigerated cooling system consists of three basic parts: a.) a cold surface, to which the electronic equipment is attached and which is at a temperature below the surroundings temperature; b.) a refrigeration system, which pumps heat from the cold temperature surface to a higher temperature surface; and c.) a hot surface, from which the heat transfer of the cold surface and that of the refrigeration power consumption are transferred to the surroundings.

Figure 1 illustrates the refrigeration system that uses a cooling fluid. The electronic equipment is mounted to the cold surface, which is at a temperature below the temperature of the surroundings. Since the cooling fluid is colder than the equipment, it picks up heat at the cold surface and transports it to the evaporator, where it is picked up by the refrigeration cycle. The refrigeration cycle consists of five main parts: an evaporator, a compressor, a condenser, a receiver, and an expansion valve. The total heat reject to the surroundings by the condenser is the sum of the heat absorbed from the cooling fluid in the evaporator and the compressor input power.

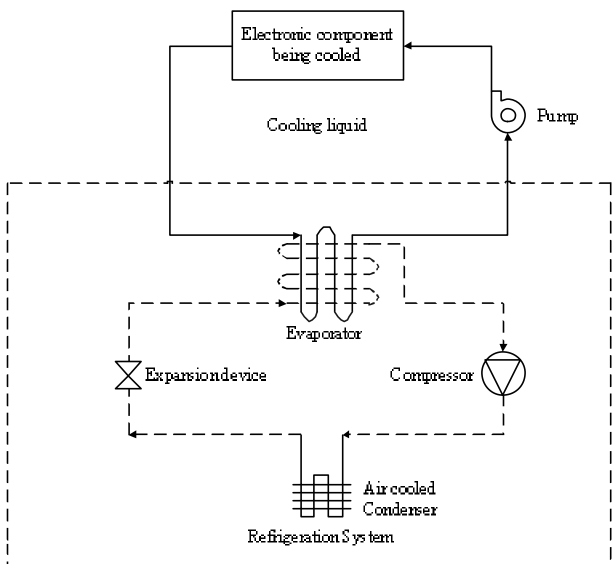

Figure 1: Refrigeration system using a cooling fluid.

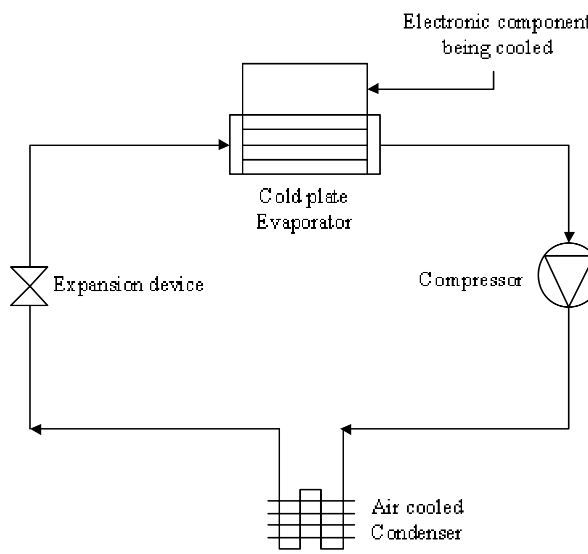

Figure 2: Refrigerated heat sink system.

A refrigerated heat sink, as shown in Figure 2, has a lower temperature at the cold surface of the electronic equipment than a refrigerated system that uses a cooling fluid. The evaporator is mounted directly to the electronic equipment and thus, the electronic temperature is colder than the surroundings temperature and the cooling system is more compact than the previous cooling system. A typical cryogenic heat sink system has a temperature significantly below 0 °C, normally at –200 °C or lower. Such temperatures can be achieved by liquid nitrogen baths. In this case, the cooled electronic components are immersed into the liquid nitrogen bath that is insulated from the surroundings by using a Dewar flask. The Dewar flask is a double wall container that is made of an insulated material such as glass. The space between the walls is evacuated to minimize heat conduction and both surfaces are coated with a shiny metal film to minimize the heat radiation from the hotter surroundings to the liquid nitrogen. The electrical wiring must be insulated to prevent heat transfer. Special care must be taken regarding the insulation technique since generally the heat dissipation by the electronic component is small comparing to the heat dissipated by the flask. Thermoelectric cooling modules operate by using the Peltier effect. Therefore, the refrigeration system is compact and requires no compressor, evaporator, condenser, and refrigerant. The thermoelectric cooler consists of a type P and type N semiconductor of bismuth telluride. A junction of these two different semiconductor materials is placed between the surface to be cooled (heat source), and the hot surface (heat sink). A DC voltage is applied across the hot surface where heat is transferred to the surroundings. The extra electrons in the N type semiconductor and the holes in the P type material are carriers that move heat from the cold to the hot junction. The heat is pumped by virtue of the Peltier effect. The rate of heat absorbed at the cold junction is pumped to the hot junction proportional to the carrier current passing through the circuit. A good thermoelectric semiconductor material such as bismuth telluride provides an easy flow of carrier to transfer heat from the cold to the hot junction and to impede the heat transfer in the opposite direction. To obtain a higher temperature difference across the cold and hot surface, the thermoelectric modules can be cascaded in series. Nevertheless, the capacity and efficiencies of the cascaded thermoelectric cooler are small.

3. COMERCIAL REFRIGERATION SYSTEMS FOR ELECTRONICS COOLING

Schmidt and Notohardjono [5] used vapor-compression refrigeration in electronic cooling of the IBM S/390 G4 CMOS (Complementary-Oxide Semiconductor) system, the first IBM system to employ refrigeration cooling. The IBM G4 server is shown in Figure 3. The blowers provide air cooling for all components of the server, except for the process module that is cooled by refrigeration. There are two modular refrigeration units (MRUs), as shown in Figure 4, providing cooling to the evaporator, which is mounted on the multi-chip module (MCM). However, only one MRU is used to cool the server during normal operation. The other MRU is a backup system for redundancy and automatically turns on when the first one fails.

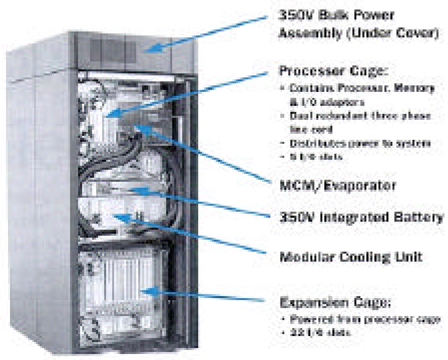

Figure 3: IBM G4 server refrigerated system [5].

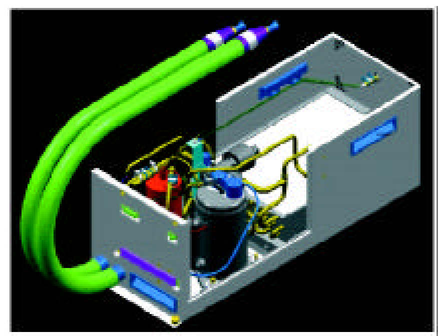

Figure 4: Refrigeration system components of modular refrigeration unit (MRU) [5].

The size of the unit is 267×267×711 mm^3 and the weight is 27 kg. The system consists of all of the refrigeration components excluding an evaporator, which is attached to the chip. Since the power supply was restricted for the entire server, a rotary compressor with brushless DC motor was chosen to minimize the startup operating current. The evaporator is an airtight metal enclosure with one open face with a butyl rubber gasket to seal and prevent moisture condensation on the evaporator. The inside of the evaporator enclosure has about 260 grams of silica gel desiccant to absorb any moisture leaking into the evaporator enclosure for the lifetime of the server. The average processor temperature for the G4 server was approximately 40 °C. This is about 35 °C lower than the air cooling system of the same design. The MRU components have been tested and reviewed to ensure highly reliable refrigeration cooling.

Peeples [6] identified possible mechanical assisted cooling methods for high power electronics in which a heat sink operates below ambient temperature. This heat sink is typically called a cold plate. Mechanical assisted cooling or active cooling system can be operated over a wide range of temperatures. However, the temperature levels that are most useful and do not provide electronic packaging problems are above 233 K (-40 °C). The three most common cold plate technologies to cool high power electronics are thermoelectric devices, chilled fluid loops, and vapor compression refrigeration. He suggested vapor compression refrigeration as the most suitable due to the following advantages: low mass flow rate, high COP, low cold plate temperatures and the ability to transport heat away from its source. He mentioned that solid-state thermoelectric devices are limited to low power applications due to their comparatively low COP. In order to achieve high power applications, the thermoelectric devices have to be cascaded in multiple stages that possess even lower COP and may require additional active cooling to remove heat from the rejection side of the devices. However, thermoelectric devices have one advantage in that there are no moving parts. A circulating chilled liquid system has been used prior to any other methods and can transport a high heat load. The most general working fluid is water. Nonetheless, there are limitations with respect to the working fluid in that it requires a relatively high mass flow rate and a secondary refrigeration system to chill the working fluid. In addition, the vapor compression refrigeration operates at a COP about three times that of thermoelectric devices and requires much less coolant mass flow rate than a chilled liquid system.

The KryoTech super G^{TM} computer with vapor compression refrigeration is shown in Figure 5. The most common refrigerants used for high power electronics cooling technology are R-404a and R-134a. Peeples [7] suggested a sub-ambient cooling to have faster switch on and off of CMOS transistors. Additionally, carrier mobility, junction leakage, sub-threshold operating characteristics and interconnect conductivity favor a lower temperature. The critical problem in electronics is water (moisture) that condensate on exposed surfaces below the dew point air temperatures. Thus, the cold exposed surfaces must be insulated and sealed to avoid condensation.

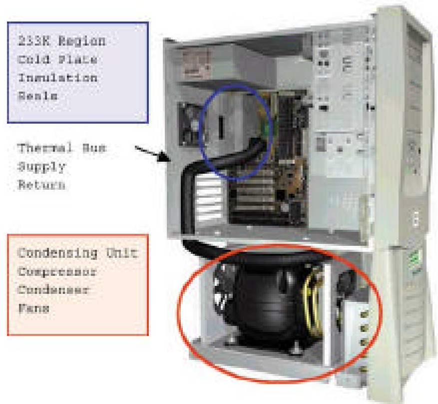

Figure 5: KryoTech super G^{TM} computer with refrigeration cooling system [6].

Nowadays, there are still not many computers that are using vapor compression cooling technology. This is mainly due to their bulky and heavy compressor, and impeccably interactive capacity control with the electronic system.

KryoTech [8] developed another miniature refrigeration cooling system with a cooling capacity of 130 W at an integrated circuit heat spreader temperature of 20 °C. A variable speed, brushless 24 volts DC rotary compressor with the size of 3.5 inch × 2.5 inch, a displacement volume of 1.8 cm^3, and a weight of 3 lb$_m$ was used to fit into the 2-U rack criteria. R-134a is the selected refrigerant with polyolester oil as the lubricant. The cooling system is composed of a copper cold plate, a rotary compressor, an aluminum micro-channel condenser with two 80 mm × 80 mm axial fans, and a capillary tube. The total dimension of the cooling unit is 16×7×2.5 in^3. Figure 6 illustrates the small-scale refrigeration cooling system. The pressure ratio as a function of compressor power input is shown in Figure 7. Figure 8 illustrates the compressor power input versus heat load of the evaporator. The COP as a function of the evaporating temperature and ambient air temperature is depicted in Figure 9.

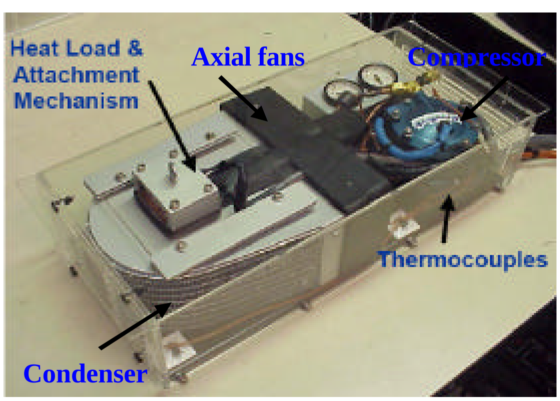

Figure 6: Small-scale refrigeration cooling system [8].

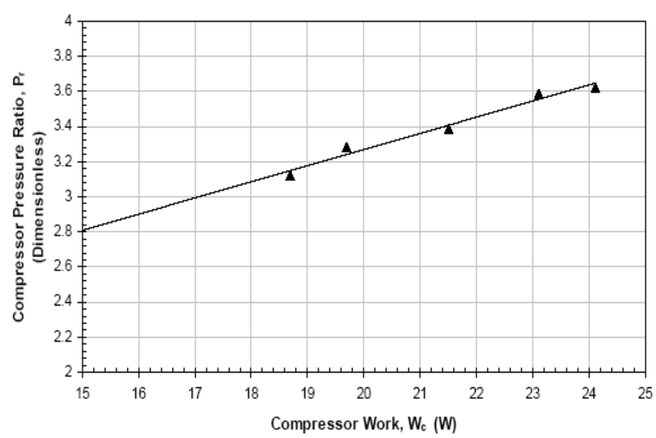

Figure 7: Compressor pressure ratio versus compressor power input [8].

In addition, the thermal resistance between the cold plate and ambient temperature, as shown in Figure 10, was calculated by:

$$q_{sa} = \frac{T_s - T_a}{Q_{load}} \tag{1}$$

Figure 11 shows the COP, evaporating temperature, and thermal resistance for various compressor speeds. The cold plate-to-ambient thermal resistance decreases with increasing compressor speed.

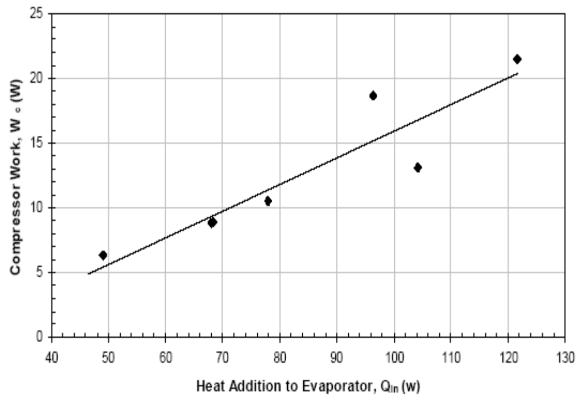

Figure 8: Compressor power input versus CPU heat load [8].

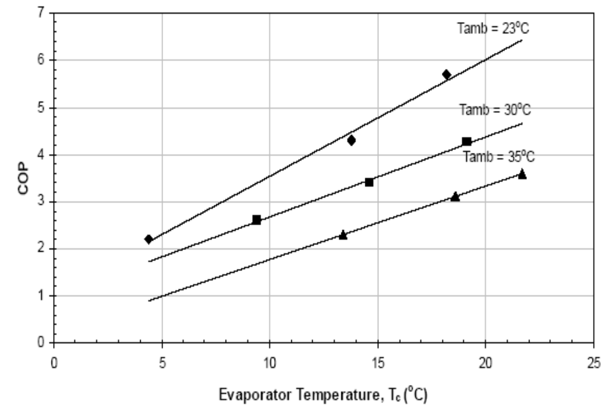

Figure 9: COP versus ambient and evaporating temperatures [8].

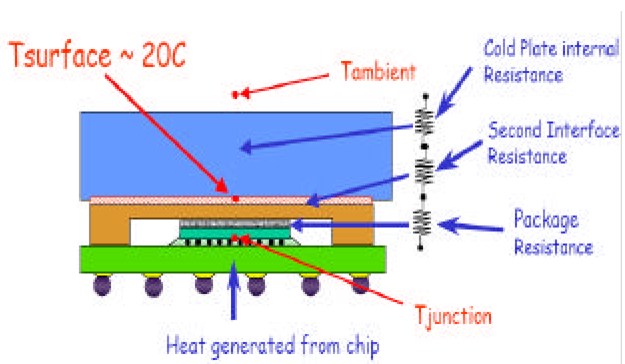

Figure 10: Cold plate-to-ambient thermal resistance [8].

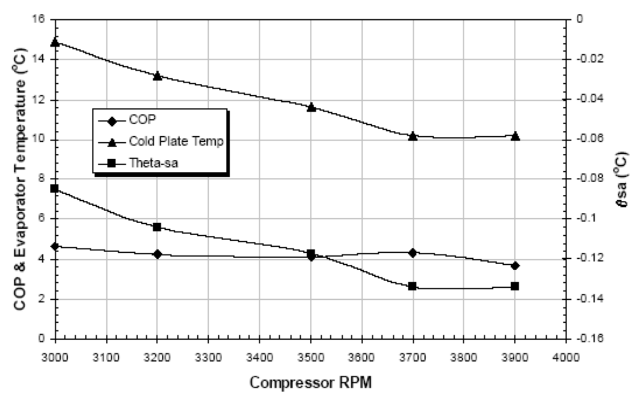

Figure 11: COP, evaporating temperature, and thermal resistance versus compressor speed [8].

4. REVIEW OF REFRIGERATION SYSTEM SIMULATION MODELS FOR ELECTRONICS COOLING

In order to analyze a miniature-scale vapor compression system for electronics cooling, the authors are currently developing a detailed system simulation model that predicts the inlet and outlet states of each component as well as the system performance. The model will be used to evaluate the performance at different operating conditions and to optimize the design of each component and the overall system based on a given set of design criteria and available space. Thus far, only two studies were found in the literature that presented refrigeration system models for electronics cooling. Both models did not include the heat spreader or chip package.

Bash [9] modeled a vapor compression refrigeration system for electronics cooling by using both thermodynamics and heat transfer to analyze and simulate the cooling system. A typical vapor compression refrigeration cooling system consists of an evaporator (which receives heat from the electronic chips), an accumulator (which prevents liquid refrigerant from entering the compressor), a compressor (which increases the vapor refrigerant pressure), a condenser (which rejects heat from the vapor compression cycle to the ambient air), and an expansion device (which decreases liquid refrigerant pressure). An alternative, hot-gas bypass valve was used to control the refrigerant flow to the evaporator and thus, to control the evaporator capacity and accommodate fluctuating chip powers, and to prevent the condensation of moisture in the air if the evaporator surface temperature drops below the dew point. The main purpose of a typical vapor compression system analysis is to calculate the thermodynamic state points of each component, the refrigerant mass flow rate, and the compressor power consumption. The cooling capacity is determined from the product of the mass flow rate and the enthalpy difference across the evaporator. The electronic chip temperature is known based on the ratio of cooling capacity to total thermal resistance of the evaporator construction. The COP of the cooling system can be computed from the cooling capacity divided by the compressor power. The refrigeration cycle analysis was conducted by using an algorithm introduced by Braun [10]. The two-phase heat transfer coefficient and the pressure drop of the evaporator were calculated by using Chen and Martinelli correlations, respectively. For the condenser, the Dittus-Boelter equation is used for single-phase heat transfer while the Chato correlation is employed for two-phase heat transfer at low Reynolds number.

In addition, Bash constructed and tested a refrigeration cooling system with a cooling capacity of 400 W at 25 °C. The evaporator heat load was varied from 210 to 400 W, and the evaporator temperature was maintained closed to 20 °C by varying the compressor speed between 63 % and 91 % according to the heat load of the evaporator using a PID control system. The simulation results were compared with experiment data. The model predicted the evaporator and condenser temperatures with a maximum error of 10 % at 400 W of evaporator heat load and with an average error of 5 % for all loads. The predictions of COP at three loads (210, 300, and 400 W) had a maximum error of 8 % at 210 W heat-load and an average error of 6 % for all loads.

A thermodynamic model of a small-scale refrigeration system for electronics cooling in computers at steady-state operating conditions was developed by Heydari [11] to evaluate the effects of the choice of refrigerant, evaporating and condensing temperatures on the system performance. The reliability, availability and serviceability (RAS) of the small-scale refrigeration system was discussed to evaluate the potential of employing such systems in high performance CPU applications, instead of the conventional air cooling system. The small-scale CPU refrigeration system is composed of a free piston linear compressor, a condenser, a capillary tube, and a cold plate evaporator. The steady-state vapor compression simulation model developed by Fischer [12] was modified to simulate the miniature CPU vapor compression cooling system. The condenser model developed by Kempaik [13] was used, which divided the condenser into desuperheating, two-phase, and subcooled regions. The heat transfer rate in each region is given by:

$$Q_{dsh} = \dot{m}_r \left(h_{cond,in} - h_{cond,g}\right) = F_{dsh} LU_{dsh} \left(T_{cond,dsh} - T_{cond,air}\right) \tag{2}$$

$$Q_{TPh} = \dot{m}_r \left(h_{cond,g} - h_{cond,f}\right) = F_{TPh} LU_{TPh} \left(T_{cond,TPh} - T_{cond,air}\right) \tag{3}$$

$$Q_{sub} = \dot{m}_r \left(h_{cond,f} - h_{cond,out}\right) = F_{sub} LU_{sub} \left(T_{cond,sub} - T_{cond,air}\right) \tag{4}$$

The products of heat transfer length and overall heat transfer coefficients, U_i , are assumed to be a function of the air volume flow rate, $\dot{V}$:

$$\frac{1}{LU_i} = b_i + \frac{c_i}{\dot{V}^{0.5}} \tag{5}$$

Equations (2) through (4) are used together with the corresponding heat transfer coefficient equations for each region to determine the constants given in equation (5) using a least square method approach. Similarly, the pressure drop in each region is obtained by:

$$\Delta P_{dsh} = \frac{\dot{m}_r^2}{A_c^2 r_{cond,g}} - \frac{\dot{m}_r^2}{A_c^2 r_{cond,in}} + \left(b + \frac{c_{dsh}}{\dot{m}_r^{0.237}}\right) L \frac{F_{dsh}}{d} \frac{\dot{m}_r^2}{2A_c^2 r_{cond,in}} \tag{6}$$

$$\Delta P_{TPh} = \frac{\dot{m}_r^2}{A_c^2 r_{TPh,f}} - \frac{\dot{m}_r^2}{A_c^2 r_{TPh,g}} + \left(b + \frac{c_{TPh}}{\dot{m}_r^{0.237}}\right) L \frac{F_{TPh}}{d} \frac{\dot{m}_r^2}{2A_c^2 r_{cond,f}} \tag{7}$$

$$\Delta P_{sub} = \frac{\dot{m}_r^2}{A_c^2 r_{cond,out}} - \frac{\dot{m}_r^2}{A_c^2 r_{cond,f}} + \left(b + \frac{c_{sub}}{\dot{m}_r^{0.25}}\right) L \frac{F_{sub}}{d} \frac{\dot{m}_r^2}{2A_c^2 r_{cond,f}} \tag{8}$$

The constants in the above three equations are solved by a least square method approach. Therefore, the total heat transfer rate and pressure drop are then calculated by:

$$Q_{cond,tot} = Q_{dsh} + Q_{TPh} + Q_{sub} \tag{9}$$

$$\Delta P_{cond,tot} = \Delta P_{dsh} + \Delta P_{TPh} + \Delta P_{sub} \tag{10}$$

The cold plate evaporator is attached to the CPU so that the heat from the CPU source is transferred to the evaporating refrigerant. Domanski's evaporator model [14] was modified to simulate the cold plate evaporator. For the refrigerant side, a multi-region flow model consisting of a two-phase flow model, an all-vapor flow model, and single phase superheated flow model were developed. The annular flow regime was assumed to exist up to a refrigerant flow quality of 0.85, the dispersed flow regime was assumed to exist between a refrigerant flow quality of 0.85 and 1.0, and the superheated flow regime was assumed to exist at a quality greater than 1.0. The heat transfer rate in each flow regime is calculated as follows:

$$Q_{evap,anl} = \dot{m}_{air} C_{p,a} \left(T_{air,i} - T_{evap,i}\right)\left[1 - \exp\left(-\frac{UA_o}{\dot{m}_{air} C_{p,a}}\right)\right] \tag{11}$$

$$Q_{evap,dp} = \dot{m}_{air} C_{p,a} \left(1 - ANNUL\right)\left(T_{air,i} - T_{evap,i}\right)\left[1 - \exp\left(-\frac{UA_o}{\dot{m}_{air} C_{p,a}}\right)\right] \tag{12}$$

$$Q_{evap} = \dot{m}_{air} C_{p,a} \left(T_{air,i} - T_{evap,i}\right)\left[1 - \exp\left(-\frac{UA_o}{\dot{m}_{air} C_{p,a}}\right)\right]\left[1 - \exp\left(-\frac{\left(1 - ANNUL - XDRY\right)\dot{m}_{air} C_{p,a}}{\dot{m}_r C_{p,r}}\right)\right] \tag{13}$$

Where the fraction of annular flow, ANNUAL, and the fraction of dispersed flow, XDRY, in the heat exchanger are calculated by

$$ANNUAL = \frac{\dot{m}_r \left(h_{r,0.85} - h_{r,i}\right)}{\dot{m}_{air} C_{p,a} \left(T_{air,i} - T_{evap,i}\right)\left[1 - \exp\left(-\frac{UA_o}{\dot{m}_{air} C_{p,a}}\right)\right]} \tag{14}$$

$$XDRY = \frac{\dot{m}_r \left(h_{r,v} - h_{r,i}\right)}{\dot{m}_{air} C_{p,a} \left(1 - ANNUAL\right)\left(T_{air,i} - T_{evap,i}\right)\left[1 - \exp\left(-\frac{UA_o}{\dot{m}_{air} C_{p,a}}\right)\right]} \tag{15}$$

In addition, the total heat transfer rate of the cold plate heat exchanger is computed by

$$Q_{evap} = UA\left(T_{evap} - T_{source}\right) \tag{16}$$

$$UA = \frac{1}{\frac{1}{\left(h_f hA\right)_R} + \frac{L}{(kA)_{CO}} + \frac{1}{(hA)_{IN}}} \tag{17}$$

In the compressor model, the refrigerant mass flow rate can be computed by knowing the volumetric efficiency, h_{vol}:

$$h_{vol} = \frac{\dot{V}_{actual}}{\dot{V}_D} \tag{18}$$

The actual power input of the compressor can be obtained by knowing isentropic, mechanical, motor efficiencies including the heat loss factor of the compressor.

An expansion device is used to control the refrigerant flow rate. A capillary tube, which is a simple and low cost device compared to other expansion or flow restriction devices, was used in the simulation. The amount of refrigerant flowing through the tube depends on the pressure differential between the evaporator and the condenser, as well as the saturation pressure of the refrigerant entering the tube. The correlation of Wolf et al. [15] is employed to calculate the refrigerant mass flow rate by using the Buckingham p theorem to find a number of relevant p -terms. Two cases of inlet conditions to the capillary tube were considered: subcooled and two-phase inlet conditions. For subcooled inlet conditions, the subcooling temperature (ΔT_{sub}) was considered to be in the range of 1 to 17 °C, and for two-phase inlet conditions, the vapor quality (x) was considered to be in the range of 0.03 to 0.25. The refrigerant mass flow rate is determined from p_8 , which is given by:

$$p_8 = Bp_1^{\exp 1} p_2^{\exp 2} p_3^{\exp 3} p_4^{\exp 4} p_5^{\exp 5} p_6^{\exp 6} p_7^{\exp 7} \tag{19}$$

Where, B is the correlation coefficient. p -terms and the exponent terms of equation (19) can be found in the original paper [10]. All fluid properties for the vapor and liquid phase are evaluated at the saturation state corresponding to the fluid temperature at the inlet of the capillary tube.

The simulation of the steady-state refrigeration system was performed by simultaneously solving a set of equations of the compressor, condenser, expansion device, and evaporator models. The essential input parameters are the suction and discharge pressures, the superheat temperature of the compressor, including the geometry of each component. EES was used to solve the set of equations and to calculate the refrigerant properties.

The simulations of the CPU spot cooling refrigeration system using the miniature linear compressor, compact air-cooled condenser, cold plate evaporator, and capillary tube were performed by assuming a junction temperature of 86 °C. The simulations were executed for various refrigerants. The results of the simulations indicated that R-134a is considered to be the favorite refrigerant for the CPU spot cooling using a vapor compression refrigeration system since R-134a has several good characteristics such as high COP, few safety concerns, environmental friendly refrigerant, as well as overall system size and cost issues, even though Ammonia and R-718 have slightly higher COPs than R-134a as depicted in Figure 12. For a fixed junction temperature and evaporator load, the COP of the system increases from about 1.7 to 3.6 as the evaporator temperature varies from about 0 to 25 °C. Likewise, for a fixed junction temperature, evaporator load, and evaporating temperature, the COP decreases from about 5.3 to 2.7 as the condensing temperature increases from about 45 to 65 °C.

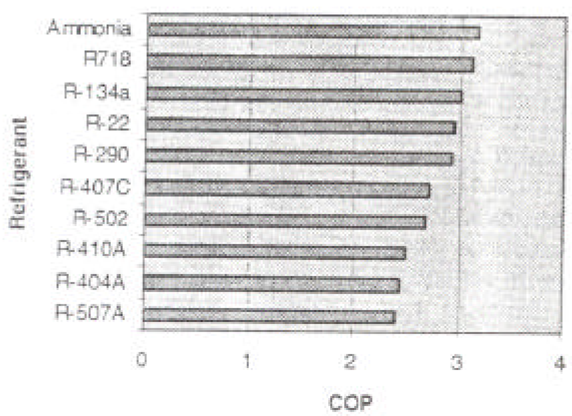

Figure 12: Effect of the choice of refrigerant on COP for given conditions [11].

5. ONGOING EFFORT

As mention earlier, the authors are part of an ongoing research effort to investigate the feasibility of a miniature-scale vapor compression system for electronics cooling, which includes the design and construction of a bread board experimental system. The target cooling capacity of the experimental system is 200 W at evaporating temperatures of 10 to 25 °C, condensing temperatures of 35 to 60°C, and ambient temperatures of 25 to 50 °C. In addition, a detailed system simulation model of miniature-scale vapor compression systems including heat spreader and chip package to predict the cooling capacity, refrigerant mass flow rate, refrigerant pressure drop of evaporator, air and refrigerant pressure drop of condenser, compressor power consumption, system COP, inlet and outlet states of each component, as well as the temperature profile of the heat spreader for various operating conditions will be developed. After model validation, the model will be used to conduct parametric studies in order to evaluate design changes and optimize the performance of MSRS for electronics cooling. The current status of the model development and experimental bread board system and the early theoretical and experimental results are presented in a companion paper [16].

6. CONCLUSIONS

The miniature-scale vapor compression refrigeration system is one of the most promising alternative cooling techniques for high heat dissipation electronics cooling. It maintains an operating chip temperature below the ambient air temperature, increases reliability and life cycle time due to a lower and constant operating temperature, and provides high system performance. An ongoing experimental and theoretical research effort will focus on optimizing the size and performance of this technology for a variety of operating conditions and a given set of design conditions.

NOMENCLATURE

A	Area	(m^2)
b, c, F	Factor or coefficient	(-)
C	Specific heat	(J/kg-K)
f_Q	Heat loss coefficient of compressor	(%)
h	Enthalpy	(J/kg-K)
k	Thermal conductivity	(W/m-K)
L	Length	(m)
$\dot{m}$	Refrigerant mass flow rate	(kg/s)
P	Pressure	(Pa)
Q	Heat loss or heat transfer rate	(W)
T	Temperature	(°C)
U	Conductance	(W/m^2-K)
$\dot{V}$	Volume flow rate	(m^3/s)
$\dot{W}$	Power consumption or power input	(W)
x	Refrigerant quality	(-)

Greek symbol

Δ	Delta	(-)
h	Efficiency	(%)
r	Density	(kg/m^3)
q	Thermal resistance	(°C /W)

Subscript

a	Ambient air or air
anal	Annular
c	Cross-sectional
co	Cold plate
comp	Compressor
cond	Condenser
dp	Dispersed flow regime
dsh	Desuperheat
D	Displacement or diameter
evap	Evaporator
f	Fluid or liquid or friction factor
g	Gas or vapor
i, in	Inlet or inner
o	Overall or outlet
R	Refrigerant
s	Surface
sub	Subcooling
tot	Total
TPh	Two-phase
vol	Volumetric

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