Modeling of MEMS-type Devices for Microprocessor Cooling

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Abstract

This paper discusses the use of MEMS-type devices for the cooling of microprocessors. Vapor compression refrigerators and circulated-coolant heat-exchange systems are analyzed. Water and commercial refrigerants are considered as working fluids. Calculations indicate that although vapor compression systems are capable of removing greater amounts of heat, circulated-coolant systems are far more practical in terms of size and power requirements. The use of a circulated-coolant system can remove up to 400 W of heat from microprocessors at 120°C to ambient air at 35°C, about three times the heat removal current heat-sink systems achieve.

Keywords: Modeling, microprocessor, refrigerant, heat exchanger, two-phase flow

1 - INTRODUCTION

Microprocessors have been traditionally cooled by attaching them to an aluminum heat sink and blowing air across the heat sink. The thermal resistance associated with this arrangement has typically been low enough to reject the heat generated in the microprocessor while keeping it at an acceptable temperature. However, since chip power densities have been rising rapidly, there is considerable interest in developing methods to remove more heat across the same temperature difference [1].

Heat sinks currently in use are capable of removing up to about 130 W of heat to ambient air at 35°C while maintaining the chip at or below a temperature of 120°C. This corresponds to a resistance of 0.65 K/W. The thermal resistance is mostly due to the contact resistance between the chip and the heat sink, and the convection resistance between the heat sink and the air outside.

Improvements in the overall heat sink resistance are severely restricted by the contact resistance. This paper discusses the use of a fluid circulated around in a closed loop to cool a chip. Using a fluid to actively cool the chip effectively eliminates the contact resistance. In addition, the fluid can be conveniently flowed over "hot spots" on the chip surface. Heat transfer coefficients associated with boiling and condensation are extremely high and result in low thermal resistances. This opens up the possibility of using a fluid-based cooling system that can be integrated with the chip and can remove far greater amounts of heat than is possible with a heat-sink-based system.

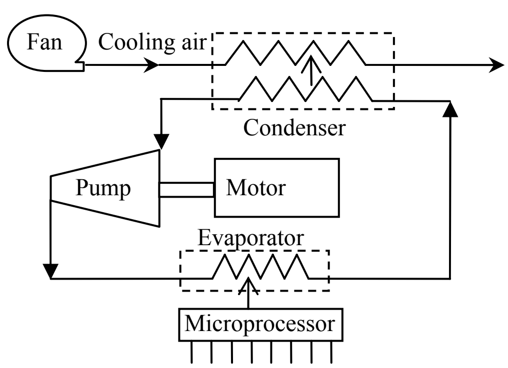

Figure 1 – A schematic of a system using a fluid pumped across a closed loop to cool a microprocessor.

In this paper, two different fluid-based configurations are discussed: 1) a simple circulation system, in which a fluid is used to convect heat away from the chip, and 2) a refrigeration system, which involves the use of a thermodynamic cycle to actively cool the chip.

2 – CIRCULATION SYSTEM

In a circulation system, the fluid is pumped in a closed loop as shown in Fig. 1. This system essentially replaces the contact and conduction resistance of a heat sink with the convection resistance of a flowing coolant. The coolant is heated by the chip in a heat exchanger and rejects the heat to ambient air in a second heat exchanger. The low thermal resistances needed to make this system viable require two-phase flow (boiling or condensation) in each of the heat-exchangers, especially since the area available for heat transfer is quite limited. These heat exchangers will be referred to as the evaporator and condenser respectively. The evaporator, pump and motor can be integrated with the chip, while the condenser can be mounted on the chip package. A similar concept is also discussed by Garimella et al [2].

Estimates of the thermal resistances of the heat exchangers are made in this section. The working fluid is assumed to be water. Other possible coolants are discussed later.

The Evaporator – A schematic of a simple evaporator design is shown in Fig. 2. The fluid flows in rectangular channels in direct contact with the surface of the chip. The total area of the chip is about 4 cm^2. The channels are about 5 mm in width and 2.5 mm in depth, and the walls of the passages are about 1 mm thick. The area available for heat transfer is assumed to be about 2.5 cm^2.

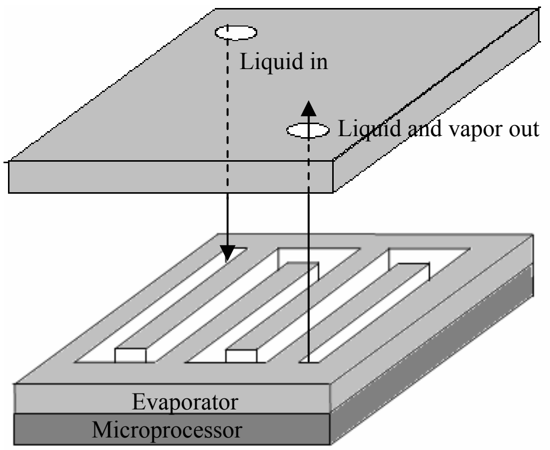

Figure 2 – Evaporator with fluid flowing in rectangular channels

The forced convection boiling heat transfer coefficient in the evaporator was calculated using [3, p.15.98-103]

$$h = \frac{k_l}{D} 0.023 \left(\frac{(1-x)\dot{m}'' D}{\mu_l} \right)^{0.8} \left(\frac{\mu_l C_{pl}}{k_l} \right)^{0.4} \left(1 + 3000 \left(\frac{q''}{\dot{m}'' h_{fg}} \right)^{0.86} \right) \quad (1)$$

where k_l, C_{pl} and μ_l are the thermal conductivity, constant-pressure specific heat and viscosity of the liquid phase, D is the diameter of the tube, x is the quality of the saturated mixture (initially assumed and later shown to be very small in this case), q'' is the heat flux as defined in [3], and $\dot{m}''$ is the mass flux through the tube. The fluid properties are calculated at a temperature of 70°C.

The heat transfer coefficient is determined to be 48100 W/m^2K for a mass flow rate of about 25 g/s. For a temperature difference of 35°C, an available area of 2.5 cm^2 and a heat transfer coefficient of about 48100 W/m^2K, the total heat transfer is about 420 W. To ensure that the flow remains two-phase throughout the evaporator, the minimum required flow rate of water is calculated from

$$\dot{Q}_{chip} = \dot{m} h_{fg} \quad (2)$$

where h_{fg} is the heat of vaporization. A heat transfer rate of 420 W and a mass flow rate of 25 g/s correspond to an enthalpy change in the evaporator of much less than h_{fg}; therefore, the water stays in the two-phase region. For this heat transfer rate and mass flow rate, the quality of the water changes from 0 at the inlet to 0.0073 at the outlet. Since there is a density change in the evaporator, a pressure difference of 0.76 bar is required to accelerate the fluid. The viscous pressure drop across the evaporator is calculated from the turbulent-flow straight-tube correlation and is determined to be 0.23 bar.

This estimate has been based on several approximations; however, as a comparison, Brunschwiler et al [4] have succeeded in removing 370 W/cm^2 from a chip at an effective heat transfer coefficient of 60000 W/m^2K, across a ΔT of 63°C, a flow rate of 40 g/s, and a pressure drop of 0.35 bar. Even though the geometry of their heat exchanger is not as simple as that assumed in this analysis, the overall dimensions are comparable.

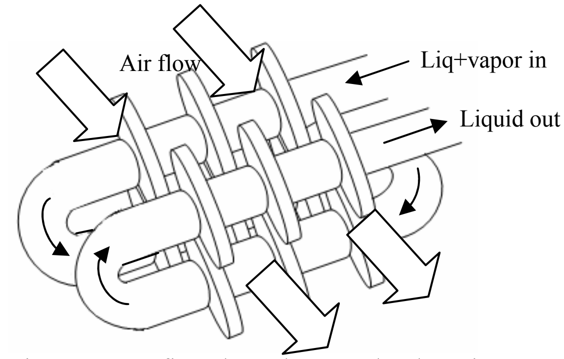

Figure 3 – A finned condenser tube, bent into a 2x2 grid, water flowing on the inside and air on the outside

The overall thermal resistance associated with the evaporator is determined to be of the order of 0.09 K/W. We now look at the condenser design.

The Condenser – The design of the condenser is quite critical since the heat transfer coefficient on the air side is expected to be quite low. Therefore the area available for heat transfer on the air side needs to be as high as possible while keeping the overall volume low. For this reason, fins are used to augment heat transfer on the air side. A possible design using a finned circular tube is shown in Fig. 3. Air flows across the tube, while the coolant condenses inside. The finned tube is about 1 m long, bent into 25 segments of 4 cm each. The tube is bent in such a way that the 25 segments are arranged in a 5x5 grid, so as to form a tube bank transverse to the direction of air flow. (Figure 3 shows a tube bent into 4 segments, arranged in a 2x2 grid.) The overall volume occupied by the condenser is 40 mm x 40 mm x 40 mm. There are a total (N) of 2000 fins, each with a thickness (t) of 0.2 mm. The tubes have an outer radius (r_o) of 1.75 mm; the fins have a radius (r_f) of 3.25 mm. The pipe thickness is 0.25 mm, which is low enough for the conduction resistance to be negligible.

The thermal resistance of a finned surface is given by [5, pp. 123-127]

$$R_{fin,array} = 1 \Big/ hA_t \left(1 - \frac{NA_f}{A_t}(1-\eta_f) \right) \quad (3)$$

where $A_f = 2\pi((r_f + t/2)^2 - r_o^2)$, h is the air-side heat transfer coefficient, k is the thermal conductivity of the pipe, $A_t = NA_f + 2\pi r_o(L-Nt)$, and η_f is the fin efficiency (calculated to be 0.9 for this case).

There is no well known correlation for the determination of the heat transfer coefficient of air flowing across a finned tube bank in terms of the geometry. Therefore, the heat transfer coefficient was calculated for two cases: 1) a flow across a bank of tubes, and 2) a flow across a flat plate (which are the two phenomena that are taking place here). For the assumed outer diameter of the tubes, a correlation for the heat transfer across a bank of tubes [5, pp.378-379] was used to get $h = 75.4V^{0.6}$ W/m^2K, where V is the air velocity in m/s. Then the correlation for flow across a flat plate [5, p.354] was used to get $h = 59.5V^{0.5}$ W/m^2K. (The fin thickness $r_f - r_o$ was used as the characteristic length.) These heat transfer coefficients were checked against experimental data [6] for finned tubing of similar dimensions. The bank-of-tubes correlation overestimated the heat transfer coefficient by about 30%, while the flat-plate correlation was about the same. Therefore, this value was used in these calculations. Assuming a heat transfer coefficient of about 75 W/m^{2}K on the outside (which requires an air velocity of 1.6 m/s), we get an external resistance of about 0.13 K/W. As a comparison, current heat sink arrangements have external convection resistances of about 0.2-0.3 K/W [1,7] .

The condensation heat transfer coefficient was calculated using [3, pp.14.36-37]

$$h = 0.026 \frac{k_l}{D} \left(\frac{\dot{m}'' \left(1 + x\left(\rho_l / \rho_g\right)^{0.5}\right) D}{\mu_l} \right)^{0.8} \mathrm{Pr}_l^{1/3} \quad (4)$$

Based on the condenser dimensions listed above, the internal area available for condensation heat transfer is about 0.0094 m^2. Calculations show that the internal convection resistance is much lower than the external resistance and can be neglected. This gives a total condenser resistance of about 0.13 K/W. The condenser pressure drop is determined to be 0.5 bar.

Overall Performance

Using the calculated thermal resistances of the evaporator (0.09 K/W) and condenser (0.13 K/W), we conclude that for a water temperature of 85°C, about 400 W of heat can be rejected to air at 35°C while maintaining the chip at 120°C. This is about three times the amount of heat rejected using a heat sink. In this configuration, all that is needed is a motor-and-pump to circulate the water through the system. For an overall pressure drop of 1.5 bar, the approximate power required is only about 5 W for a pump efficiency of 75%.

Other Working Fluids

The calculations in the previous sections were done using water as the working fluid. Commercial HFC refrigerants such as R-124, R-245fa/ca and R-236ea/fa, as well as Flourinert cooling fluids such as FC-77, FC-43 and FC-3283 were also considered. However, their thermal conductivities are about 10 times less than that of water, and their heats of vaporization are also about 10-20 times less than that of water [8,9]. These fluids end up having a much larger condensation resistance, and therefore require a much larger condenser. CO_2 was also considered as a working fluid. However, its critical temperature of about 31°C precludes the possibility of two-phase flow. In this application, the thermal performance of water was superior to all the other fluids considered.

In conclusion, a circulation system involving the pumping of water in a saturated state can greatly increase the heat removal rate.

3 – REFRIGERATION

We now look at whether there are additional benefits to running the fluid in a refrigeration cycle that can be

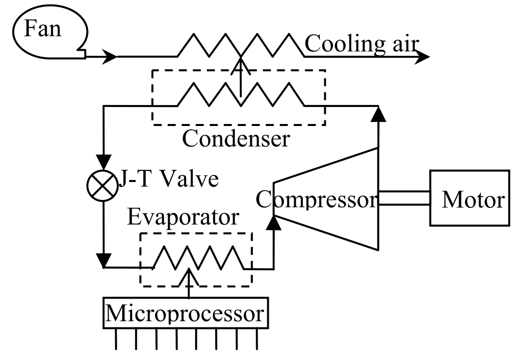

Figure 4 – Schematic of a vapor-compression refrigerator.

integrated with the chip. The refrigerator uses a work input to pump heat out of the chip.

A vapor-compression cycle (Fig. 4) with phase-change processes in the evaporator and condenser is an obvious choice, because of its high-effectiveness heat exchangers. Since several refrigerants have a critical temperature of around 100°C, the other option considered was to have the fluid go supercritical in the compressor. However, the relatively poor heat transfer characteristics in the supercritical region, away from the critical point, result in an ineffective condenser. Therefore, only a regular vapor-compression cycle was considered.

Refrigerator model

A simple thermodynamic model of a refrigeration cooling system was developed. A schematic is shown in Fig. 5. The refrigerator operates between a cold temperature T_C and a hot temperature T_H. The difference between the hot and cold temperatures is ΔT. The resistance to heat transfer between the chip surface and the fluid in the evaporator is R_1. The resistance to heat transfer between the fluid in the condenser and the air is R_2. The temperature of the chip is $T_{chip.}$ The refrigerator has a Coefficient of Performance (COP) of η times the Carnot COP.

Using the First and Second Laws of Thermodynamics, the definition of COP, the definition of thermal resistance, and the following normalised variables,

$$T_{air}^* = \frac{T_{air}}{T_{chip}}, \ \Delta T^* = \frac{\Delta T}{T_{chip}}, \ Q^* = \frac{R_2 \dot{Q}_{chip}}{T_{chip}}, R^* = \frac{R_2}{R_1}$$

we get an equation which can be solved to determine the amount of heat removed as a function of the parameters η, ΔT, R_1 and R_2.

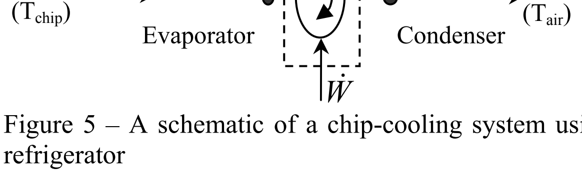

Figure 5 – A schematic of a chip-cooling system using a refrigerator

$$1-\frac{1}{\eta}\frac{\Delta T^{*}Q^{*}R^{*}}{1-Q^{*}}=Q^{*}(1+R^{*})+T_{air}^{*}-\Delta T^{*} \tag{5}$$

In the limiting case of ΔT=0, the model reduces to the circulation system model. In the previous section, we made estimates of the evaporator and condenser resistances, which can be used as values for R_1 and R_2 in the model.

The results underscore the need for high efficiency refrigerators. The heat removal rate is plotted as a function of the temperature difference across the refrigerator (ΔT) in Fig. 6, for two different refrigerator efficiencies (η = 0.1 and η = 0.3). With η = 0.1, the heat removal rate drops with ΔT, indicating that the addition of a refrigerator actually results in less heat rejection than with a circulation system with the same thermal resistances R_1 and R_2. On the other hand, a refrigerator with η = 0.3 does remove more heat than the circulation system.

The MIT micro-engine group has designed compressors with projected efficiencies of about 60% [10,11]. This immediately limits the actual COP to below 60% of the Carnot COP. Factoring in other losses, it is reasonable to expect overall COPs of no more than 30% of the Carnot COP. Figure 7 shows the heat removed from the chip and the work input required as a function of ΔT. It is apparent that refrigeration increases the heat removal rate; however, this comes at the cost of a large work input. Only 5 W of pump work are required to remove 390 W of heat using the circulation system. Using a refrigerator, 80 W of work are required to remove 430 W of heat, and 400 W of work are required to remove 510 W of heat. The benefits of using a refrigeration cycle are limited: as the work increases, the compressor and motor will get bigger, making it increasingly harder to integrate them with the chip. In addition, the length of the region in the condenser with superheated vapor will increase as the heat rejection temperature increases. This will necessitate much larger condensers to maintain the overall effectiveness, since superheated vapor has a much lower heat transfer coefficient. A small ΔT (10-20°C) refrigerator is more practical, but does not provide that much more of a benefit over the circulation system, while needing the additional components (a compressor and a JT valve).

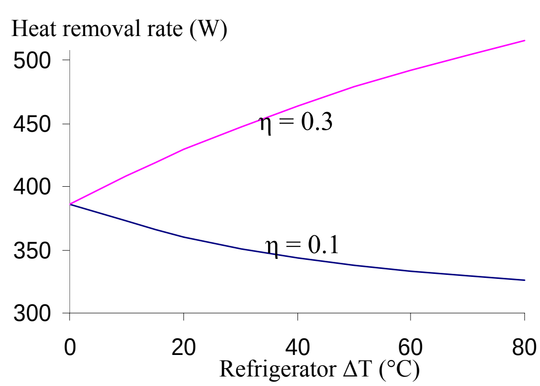

Figure 6 – Heat removal rate as a function of ΔT, for two different values of η (R_1=0.09 W/K, R_2=0.13 W/K)

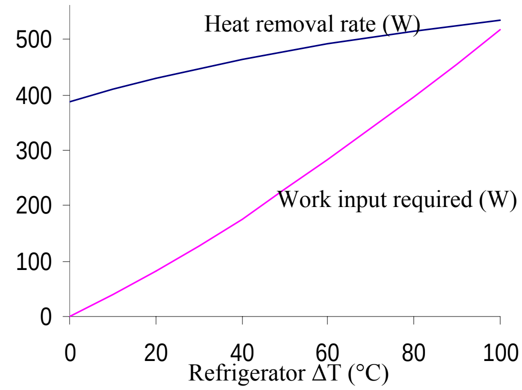

Figure 7 – The chip heat removal rate and the work input for the refrigerator (η=0.3), as a function of ΔT

Some HFC refrigerants with relatively high critical temperatures (R-245ca, R-245fa, R-236ea) were also considered as working fluids, but their heat transfer characteristics are inferior to those of water. Therefore, from a thermodynamic perspective, they are not as attractive as water.

4 - CONCLUSIONS

It is possible to increase the heat removal rate from a chip substantially by replacing the heat-sink arrangement with a circulation system integrated with the chip, in which a fluid is pumped around a closed loop. From a purely thermal standpoint, water is the best choice of fluid for this application among all the refrigerants considered. On the other hand, a refrigeration system appears impractical at current component performance because of the scale of compressors, motors and condensers required to have any benefit over the circulation system.

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