Cooling Performance of a Two-Phase (Vapor-Liquid) Flow-Cooling System

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Abstract

Cooling capability is measured for a two-phase (vapor-liquid) flow-cooling system in terms of its thermal resistance and cooling power. The performance is compared between systems with and without a pump that drives the flow of a dielectric working fluid with a low boiling point. The geometry of the boiling chamber is first examined for each flow configuration to ensure the best possible performance. The results show that using a pump does not always result in higher power consumption than that of a thermal siphon system and the driving force generated by the pump augments phase-change heat transfer thereby reducing the fan power to minimize the overall power consumption. The present results show that the flow-boiling system has a high cooling capability using a small amount of cooling power compared with the thermal siphon.

Keywords: Phase Change Cooling, Two-phase Flow, Power Consumption, Energy Efficiency, Multiple Heat Sources

1. Introduction

Reducing the thermal resistance required to cool information and communication technology (ICT) equipment is important in order to reduce the power consumption of cooling systems such as fans and air conditioners.

A thermal siphon type of phase-change cooling system has the potential to minimize the case-to-ambient thermal resistance.[1] An effective condenser of the siphon returns low temperature refrigerant to a boiling chamber, and boiling heat transfer has an aspect of lowering the thermal resistance while increasing heat transfer per unit surface area, or the heat flux.[2, 3]

A two-phase flow-cooling system, on the other hand, has the heat transfer effect of the forced convection in addition to the evaporative heat transfer of a thermal siphon. Harirchian and Garimella systematically investigated the relation between the mass flow rate of the dielectric refrigerant and the heat transfer, and regimes of the boiling phenomena by pumping and boiling the refrigerant in micro-channels.[4, 5] The coolant flow is driven by a pump to provide a convective effect on the boiling surface, in addition to the heat transfer by boiling bubbles.

The present study evaluates the cooling capability of a two-phase flow-cooling system and its potential for minimizing power consumption. We have chosen two cooling systems, one to cool a single power device and the other to cool two power devices, as they represent typical ICT equipment.

2. Experimental Setup

The experimental tool of the thermal siphon shown in Fig. 1a consists mainly of a boiling chamber and a con-
The thermal siphon uses a dielectric refrigerant with a low boiling point (Novec 649 from 3M, boiling point 49°C at 1atm).[6] The liquid is evaporated in the chamber by heating the bottom of the chamber with a heater. The refrigerant vapor is cooled in the condenser, and the condensed liquid flows downward due to gravity. Therefore, the refrigerant circulates in the experimental tool.

To stimulate the evaporation of the refrigerant, the inner pressure of the phase-change cooling system is adjusted to the saturated vapor pressure with a vacuum pump.

Figure 1b shows the boiling heat sink image in the boiling chamber. The combination of fin pitch (p) and fin thickness (t) for the heat sink in the chamber has been reduced to the following three types: p/t = 2 mm/1 mm, p/t = 1.5 mm/0.75 mm, and p/t = 1 mm/0.5 mm. The height of the fins is 10 mm.

The gap between the tips of the fins and the ceiling of the chamber is 0.5 mm for the cases with a pump in order to force the flow of the refrigerant between the fins.

A pump and a tank to store the refrigerant are installed between the inflow entrance of the chamber and the outlet of the condenser when the flow boiling cooling system is evaluated.

Experimental cases are run with the following volumetric flow rates by controlling the power input to the pump: 158, 331 and 441 ml/min.

Thermocouples are installed to the condensate outlet and the vapor inflow entrance of the condenser as well as the point of contact of the heater and chamber, and another sensor measures the ambient temperature. The same fan and the condenser are used for all cases reported in the present study.

3. Results and Discussion

3.1 Cooling system for a single power device

Figure 2 shows the case-to-ambient thermal resistance, $R_{ca}$, of the flow boiling with the pump and that of the thermal siphon when the cooling system with three types of fin pitches dissipates 100 W.

The thermal resistance, $R_{ca}$, is calculated from

$$R_{ca} = \frac{(T_c - T_a)}{Q},$$

where $T_c$ is the temperature of the base of the boiling chamber; $T_a$ is the temperature of the ambient air; and Q is the heat dissipated by the heater. The flow rate of the pump is set to 331 ml/min for this part of the study. In order to compare the effects of the fin geometry and pumped flow on the cooling system, the same condenser and fixed fan power are applied.

As shown in Fig. 2, the thermal siphon shows only a small variation in $R_{ca}$ with the fin pitches used. On the other hand, the flow boiling takes a value of $R_{ca}$ clearly smaller at $p = 1$ mm than the values for the other fin pitches.

In the case of the forced convection, the heat convection can be improved by adding forced flow. Because the pressure applied to the liquid prevents the back-flow caused by the vaporization and generates forced convection in one direction only. This effect might particularly appear when the fin pitch is less than a certain value as shown in this experiment.

In the following experiments, the fin pitch for the thermal siphon is set to $p = 2$ mm, and that for the flow boiling is set to $p = 1$ mm.

Figure 3 shows the relation between the cooling electric power and the heater power. The total cooling power is the sum of the pump and the fan, when adjusted to keep $T_c - T_a = 50^\circ C$ in the three systems (water-cooling, thermal siphon, and flow boiling systems). The data are taken at different fan power settings with a fixed pump power. The water cooling system, in which the phase change does not occur, uses the same chamber as the flow boiling system.
The power consumption of the pump is 0.48 W for a flow rate of 158 ml/min and 4.56 W for 441 ml/min. For the water-cooling and flow-boiling systems, the cooling electric power does not become 0 W because the driving force on the liquid is necessary.

In the water-cooling system, the driving electric power of the pump increases significantly though the cooling capability improves with an increase of the flow rate. The maximum value of the heater power that can be cooled is reduced as the compensation when the cooling electric power is suppressed.

When some cooling electric power is used, the flow-boiling system may show the highest cooling performance of the three methods. In other words, when the same heat source is cooled, the flow-boiling system can reduce the cooling electric power most effectively, and it becomes a means of energy conservation.

Figure 4a shows the plots of the thermal resistance, $R_{ca}$, of flow boiling by heater power while changing the flow rate.

As shown in Fig. 4a, for each heater power, when the flow rate is low, $R_{ca}$ becomes lower.

Figure 4b shows the plots of the thermal resistance, $R_{cw}$, of the boiling area.

$R_{cw} = (T_c - T_{out})/Q$,  \hspace{1cm} (2)

where $T_{out}$ is the temperature of the condensate outlet.

Figure 4b indicates that increasing the flow rate does not result in a notable improvement in $R_{cw}$. In general, a higher heat transfer coefficient is expected with an increase in flow rate. The present results show that an increase in the flow rate does not result in a major improvement of the performance of flow boiling. This suggests a possibility that the volume expansion by the evaporation of the refrigerant contributes to the improvement of heat transfer. The growing bubbles push the refrigerant downstream. The pump provides this fixed flow direction. This pushed flow increases the departure of boiling bubbles and augments the heat transfer. It enables a higher cooling performance than the thermal siphon.

It is known that the thermal resistance increases with decreasing flow rate in water-cooling systems. In this study, the $R_{ca}$ of the water cooling for the flow rate of 441 ml/min increases about 30% compared with that for 158 ml/min. As for the flow boiling system, the cooling capability of the entire system improves when the flow rate is low. This is because the cooling performance of the boiling chamber does not deteriorate with decreasing flow rate. Therefore, the pumping power may be reduced without influencing the cooling performance.

Figure 5 shows the $R_{ca}$ of the flow-boiling and thermal-siphon systems for 100 W of heater power. The horizontal axis is the cooling electric power. When the cooling electric power of the flow-boiling system is reduced, for example, the cooling capabilities remain higher than that of the thermal siphon.
3.2 Cooling system for two power devices

A two-phase flow-cooling system having two boiling chambers connected in series, shown in Fig. 6, was constructed and evaluated. The fin pitch \( p \) and fin thickness \( t \) of Chamber 1 are \( p/t = 1 \text{ mm/0.5 mm} \), and those of Chamber 2 are \( p/t = 1.5 \text{ mm/0.75 mm} \). The volumetric flow rate is 158 ml/min. \( T_{c1} \) and \( T_{c2} \) are the temperatures measured at the base of Boiling Chambers 1 and 2, respectively. \( T_{1\text{ in}} \) and \( T_{2\text{ in}} \) are the temperatures of the refrigerant flowing into the respective boiling chambers. Table 1 shows the \( R_{cw} \) of the two chambers, and the difference between the inlet temperatures \( \Delta T_{\text{in}} = T_{2\text{ in}} - T_{1\text{ in}} \) for combinations of heater power, where \( R_{cw} \) is calculated from

\[
R_{cw1} = \frac{ (T_{c1} - T_{1\text{ in}}) }{ Q } ,
\]

\[
R_{cw2} = \frac{ (T_{c2} - T_{2\text{ in}}) }{ Q } .
\]

Table 2 shows the \( R_{cw} \) when each chamber is connected as the only boiling chamber in the cooling system.

The thermal resistance of Chamber 1, \( R_{cw1} \), is greater in the two-chamber system (0.154°C/W) than in the single chamber system (0.119°C/W). \( R_{cw2} \), on the other hand, shows little change. These are due perhaps to the fact that the downstream chamber acts as flow resistance, causing the saturated vapor pressure inside chamber 1 to be greater than when it is operated in a single-chamber system. As a result, \( R_{cw1} \) is increased in the two-chamber system. Chamber 2 has no further downstream resistance, and it does not experience any increase in its \( R_{cw} \).

The value of \( R_{cw2} \) is reduced to 0.161°C/W when heater power is applied to both of the two chambers. A jet-like flow entering chamber 2 is observed during the experiment. The flow of this liquid-vapor mixture is considered to augment the convection on the surfaces of the fins because of the high velocity of the flow. This is similar to what occurs inside the nozzle of an inkjet printer, where the liquid-vapor mixture is accelerated through a narrow channel.

The temperature rise \( \Delta T_{\text{in}} \) of the refrigerant is 2.2°C for this phase-change system, compared to the estimated counterpart of about 9.2°C when 100 W heater power is cooled in the water-cooling system with 158 ml/min flow rate. This 9.2°C is calculated from

\[
\Delta T_{\text{in}} = \frac{ Q }{ C_p \rho v } ,
\]

where \( C_p \) is the specific heat of the fluid; \( \rho \) is the fluid density; and \( v \) is the flow rate of the fluid.

Therefore, when decreasing the flow rate in order to reduce the pump power, high temperature water flows into chamber 2, and the convection effect decreases. As a result, the cooling performance of the water-cooling system is significantly dependent on the flow rate, or the pump power. The two-phase system has, on the other hand, potential for limiting pumping power as well as fan power.
4. Conclusion

We have evaluated the cooling capability of a two-phase flow-cooling system and have compared its performance with that of a thermal siphon. Our results show that the flow-boiling system may be more energy efficient than the thermal siphon despite the power consumption of the pump. This is probably because the volume expansion of the boiling refrigerant contributes considerably to the cooling capability compared to that of the mass flow provided by the pump. This means that the driving electric power of the pump can be minimized while maintaining the cooling capability even if the flow rate is greater than the evaporation speed of the refrigerant. Moreover, when the same amount of heat is transferred, flow boiling reduces the rotating speed of the fan and can reduce noise and the cooling electric power consumption because the cooling capabilities are higher than the thermal siphon. When cooling a higher heater power value, the effect is more remarkable. The electric power of the fan can be reduced more than the driving power of the pump. The total cooling power may then be reduced more than the fan power may be for the thermal siphon.

The two-phase flow-cooling system with two boiling chambers in series shows that $R_{cw}$ does not increase with decreasing flow rate and that the temperature rise of the refrigerant passing the boiling chambers is smaller than that of a water-cooling system. Therefore, the two-phase flow-cooling system for cooling multiple heat sources has the potential to achieve better performance than the water-cooling system.

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References