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Experimental Research about the Heat Transfer Characters on the Two-stage Loop Thermosyphon System

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ABSTRACT

This paper reports on the construction of a two-stage loop thermosyphon cooling system and the analysis of its heat transfer mechanisms. The temperature and heat resistance distributions under various liquid filling ratios and heat loads are summarized. Further, the heat transfer boundary of the system and conditions for the two operating limits (dry-out and immersion) were determined. Finally, the heat transfer paths obtained under standard and immersion-limit operating conditions and the characteristics of coolants were analyzed and compared.

Keywords: Chip cooling, loop thermosyphon, liquid cooling

NONMENCLATURE

Abbreviations	
T	Temperature (°C)
Q	Heat transfer rate (W)
R	Thermal resistance (K/W)

1. INTRODUCTION

As computer capacity rapidly develops, the power consumption of electronic chips is increasing significantly. This will cause high internal temperature and irreversible damage to the CPU[1]. More than 55% of electronic breakdowns are caused by overheating [2]. Therefore, reliable chip-cooling technologies must be developed to ensure system stability.

Heat pipes are one of the most efficient liquid cooling technologies due to high thermal conductivities and simple structures[3][4]. Wang et al. [5]found that the thermal conductivity and thermal homogenization of the heat-pipe radiator are superior to a fin cooling system and in a CPU heat-pipe cooling system.

Although thermosyphon technology has been applied to CPU cooling, the majority of research conducted thus far has been restricted to loop thermosyphon structures featuring a single evaporator and condenser through which coolant evaporates and condenses completely without excessive flow. This paper reports on the analysis of a single condenser corresponding to multiple evaporators, in which the amount of coolant is bound to be excessive. The study focused on the cooling regulation and heat transfer limits of the loop thermosyphon system in series form, and results will be introduced in details.

2. EXPERIMENTAL SETUP

2.1 Two-stage Loop Thermosyphon System

To analyze the cooling process across multiple analog chips, a two-stage loop thermosyphon cooling system was constructed and R134a was used as the coolant. The schematic diagram of the cooling system is shown in Figure 1.

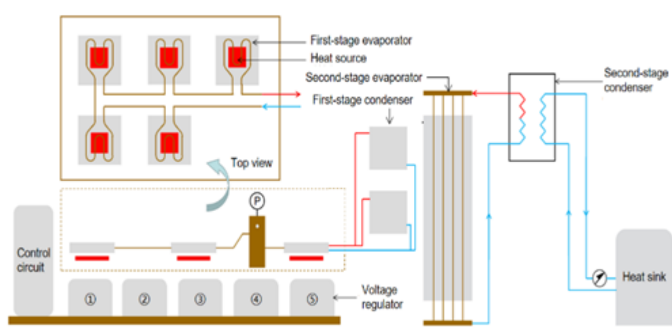

Fig. 1 Schematic of loop thermosyphon cooling system

The heat source of the system was comprised of five cooper blocks containing heaters, and the power of each block could be adjusted. The first-stage evaporators were fitted tightly onto the copper blocks and interconnected with plastic tubes, which were transparent to enable clear observation of the coolant flow patterns. The first-stage evaporators absorbed heat from the copper blocks and transported it to the first-stage condenser, and then transported heat to the second-stage evaporator. Heat was then transferred to the heat sink through the second-stage condenser, which was a plate heat exchanger situated at a relatively higher position. Water with an inlet temperature of 20℃ was used in the heat sink. The temperature difference between the water at the inlet and the outlet was set to approximately 5℃.

To reduce heat loss during the experiment, the first and second-stage evaporators and condensers were wrapped in thermal insulation. A liquid storage tank was placed between the evaporator inlet and the condenser outlet to prevent the coolant drying out or refluxing.

2.2 Experiment Method

The loop thermosyphon system was equipped with apparatus to measure temperature, pressure, and flow. Thermocouples were placed at key positions within the system, as shown in Figure 2. According to calibration, the error range of the thermocouples was ±0.1 ℃. A pressure sensor was installed on top of the liquid storage tank to ensure the system would work within the permitted pressure range and under stable operating conditions. The measurement accuracy of the pressure sensor was 0.2%. Additionally, a flow meter was installed at the heat sink outlet to measure the water flow.

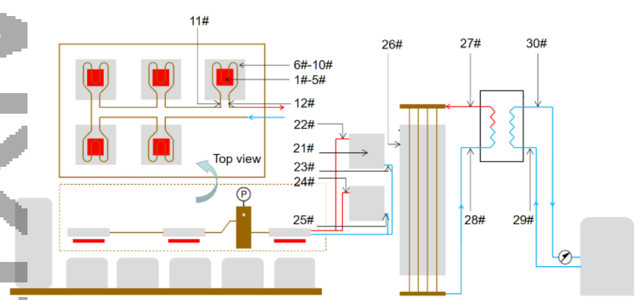

Fig. 2 Positions of thermocouple measuring points

To reproduce a variety of operating conditions, the power of each heating block could be adjusted from 20 to 100 W, thus providing a total power range of 100–500 W. Finally, the liquid filling ratio of the first-stage of the loop thermosyphon system (as defined in the following section) could be adjusted from 40% to 100%.

3. EXPERIMENTAL RESULT

3.1 Factor Definitions

The first stage system operates horizontally, so the liquid filling ratio can be defined as

$$\theta_l = \frac{M_i}{M_o} \times 100\% \quad (1)$$

where M_i is the mass of coolant charged into the first stage system and M_o is the mass of coolant required to completely fill the system. The liquid filling ratio of the second stage system was 100%, and all references to the filling ratio in this paper relate to the first stage system.

The total thermal resistance of the cooling system is defined as

$$R = \frac{T_e - T_c}{Q} \quad (2)$$

where T_e is the temperature of the heat source, which corresponds to the mean temperature of the measuring points 1#–5# (as shown in Figure 2); T_c is the temperature of the water as recorded at measuring point 29#; and Q is the amount of heat transferred by the cooling system.

A proportion of the generated heat was potentially dissipated due to ambient temperature variations. The thermal equilibrium under arbitrary operating conditions is defined as

$$\theta_h = \frac{P_w}{P_e} \times 100\% \quad (3)$$

where P_w is the power consumed by the water-based section of the loop thermosyphon system and P_e is the total power generated by the heat source.

3.2 Temperature Distribution

Temperature distributions were obtained under the operating conditions of: a liquid filling ratio of 60%, a total heat load of 100W-500W, an ambient temperature of 15℃, and a water inlet temperature of 20℃. The obtained thermal equilibrium data are shown in Figure 3.

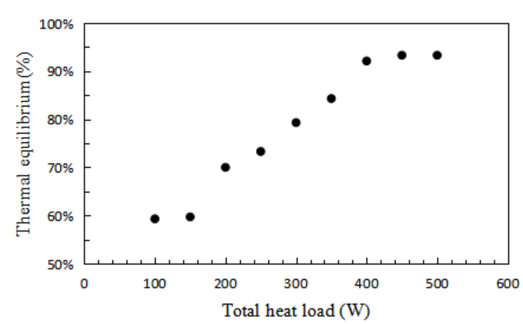

Fig. 3 Thermal equilibrium in different total heat load

As the heating power was increased, the heat transfer capacity of the water-based section of the system exceeded the heat dissipation, and therefore the thermal equilibrium gradually increased. The experimental data were reliable only when the thermal equilibrium was greater than 70%.

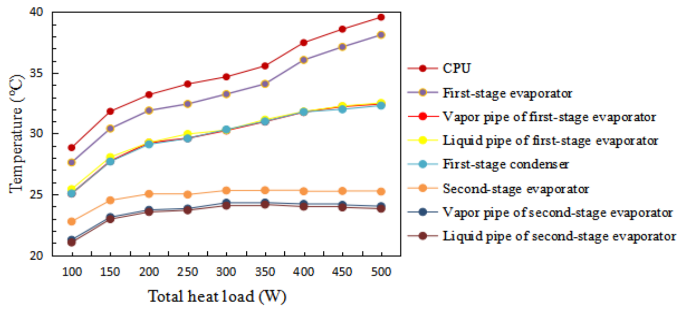

Fig. 4 Temperature distribution under different heat load

Figure 4 shows the temperature distribution with increasing total heat load for a liquid filling ratio of 60%. Total heat loads of less than 200 W produced thermal equilibria of less than 70%; therefore, these experimental data were not used in the subsequent analysis. As the heat load was increased, the temperature of the equipment in the first stage of the loop thermosyphon system increased approximately linearly, while the temperature of the equipment in the second stage remained relatively consistent due to the constant liquid filling ratio of the second loop.

3.3 Thermal Resistance Analysis

Data for thermal resistance analysis were obtained under the following conditions: a liquid filling ratio of between 40% and 100%, a total heat load of 100–500W, an ambient temperature of 15 ℃, and a water inlet temperature of 20 ℃. The temperature distributions corresponding to various operating conditions and the water flow were measured and recorded.

The results indicate that when the liquid filling ratio was between 40% and 50%, the system could not operate as intended. It should be noted that operating conditions with thermal equilibria of less than 70% have been omitted. System performance was satisfactory when the liquid filling ratio was increased to 60%, and the temperature of the heat source increased slowly with increasing total heat load. When the liquid filling ratio was increased to 80% and the total heat load was increased to 300 W, the temperature of the heat source approached the upper limit of the permitted temperature range. A liquid filling ratio of greater than 90% resulted in an excess of coolant in the first-stage of the heat-pipe system and a soaked condenser, which caused an increase in pressure and a deterioration in heat transfer. Therefore, the operation of the loop thermosyphon cooling system was adversely affected by both insufficient and excessive liquid filling ratios.

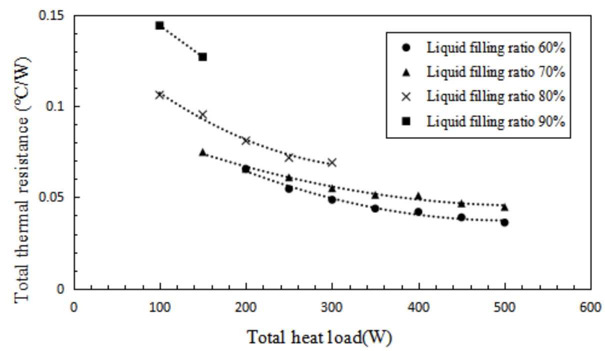

Fig. 5 Total heat load with different heat load and liquid filling ratio

The total thermal resistance was one of the key factors for measuring the heat transfer capacity of the loop thermosyphon system, and Figure 5 shows the total thermal resistance of the system under a variety of operating conditions. Under a constant liquid filling ratio, increasing the power of the heat source resulted in corresponding decreases in the total thermal resistance. At the liquid filling ratio of 60%, the heat source temperature is the lowest under the same heat load, while the total thermal resistance tends to be the smallest. Therefore, 60% was the optimal liquid filling ratio for the loop thermosyphon system.

4. HEAT TRANSFER LIMIT ANALYSIS

4.1 Heat Transfer Limit

Figure 6 shows the relationship between the liquid filling ratio and the maximum cooling capacity of the loop thermosyphon system

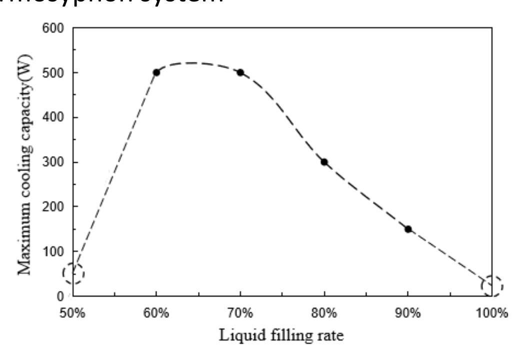

Fig.6 Heat transfer boundary of the loop thermosyphon system

The first stage of the cooling system is a gravitational thermosyphon. The coolant in the evaporator absorbs heat, partially vaporizes, and transfers heat to the condenser. The evaporated coolant in the condenser will then transfer heat onto the next stage. In the experiment, the coolant had three states: a superheated steam, a two-phase flow, and a subcooled liquid. The optimum operating state for the coolant is the two-phase flow, in which there are few temperature variations during heat transfer. If the coolant is subcooled or superheated, the flow patterns in the evaporator and condenser will not match, and the driving temperature difference between the heat source and the heat sink will increase.

4.2 Dry-out Limit

When the coolant is superheated steam, it will gradually dry out and the temperature inside the tube will increase rapidly. This is known as the dry-out limit, and generally occurs in operating conditions with a low liquid filling ratio and a high heat load. In such operating conditions, the lower part of the tube contains liquid coolant and the upper part contains vaporized coolant. The liquid coolant gradually evaporates in the direction of heat transfer, and after passing through a given number of evaporators will vaporize completely. This causes wet steam to be in direct contact with the inner tube wall. At this point, the heat transfer process deteriorates and the tube wall temperature increases rapidly. The wet steam continues to absorb heat as it passes through the remaining evaporators and becomes dry steam. Further, the vaporized coolant will be significantly superheated, which will cause a rapid rise in the temperature of the heat source that will ultimately exceed the upper safety limit.

The pressure of the first stage of the system and the temperatures at the inlets and outlets of the evaporators were recorded by a pressure sensor and thermocouples, respectively. Under standard operating conditions, the coolant flows in a two-phase state and the saturation temperature corresponding to the system pressure should be equal to the temperature of the coolant at the outlet of the evaporator. As the coolant dries out, the heat transfer will deteriorate, and liquid film cannot coat the tube wall. When this occurs, the temperature at the outlet of the evaporator will be greater than the saturation temperature, and this temperature difference is defined as the degree of superheating. Therefore, the dry-out limit should be avoided either by increasing the liquid filling ratio or reducing the total heat load.

4.3 Immersion Limit

If the liquid filling ratio increase to be larger than 70%, an immersion limit will occur. Under these conditions, the liquid coolant will immerse an increasing proportion of the heat transfer area of the condenser, and therefore the proportion of vaporized coolant that condenses will diminish. This causes a decrease in the rate of evaporation required to maintain the balance of the cooling system, and thus its heat transfer capacity will decline.

Under immersion limit conditions, the temperature of the coolant in the vapor pipe of the condenser should be equal to the saturation temperature of the corresponding system pressure. The temperature of coolant in the liquid tube was far lower than the saturation temperature, and therefore the coolant in the liquid pipe was deemed to be subcooled. Furthermore, the temperature of the coolant in the liquid pipe of the condenser should be equal to the saturation temperature of the dry-out limit when the vapor side is superheated.

To study the characteristics of the immersion limit, the temperature distributions of two sets of operating conditions were compared: a liquid filling ratio of 70% with a total heat load of 150 W (representing standard operating conditions), and a liquid filling ratio of 90% with a total heat load of 150 W (representing immersion-limit operating conditions). Figure 7 compares the heat transfer paths of the two sets of operating conditions.

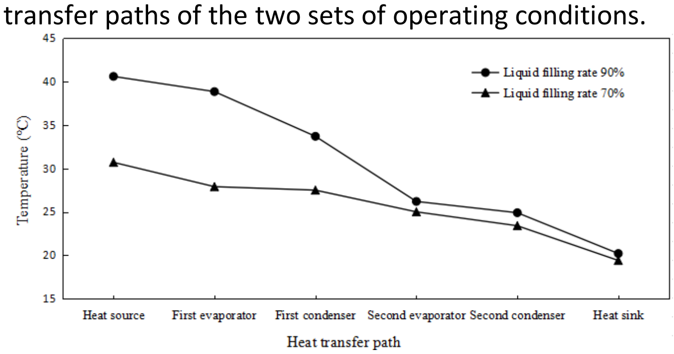

Fig. 7 Comparison of heat transfer path

Figure 7 illustrates the significant differences in the temperature distributions of the two heat transfer paths during the first stage of the loop thermosyphon system. At a liquid filling ratio of 70%, the temperature difference between the heat source and the first-stage condenser was 2.2℃, while a liquid filling ratio of 90% produced a temperature difference between these two components of 7℃. The reason behind these observations is that the excessive liquid filling ratio resulted in a relatively higher system pressure that impeded the coolant vaporization required to remove the heat generated by the heat source. Therefore, heat accumulated in the heat source, which caused a significant increase in temperature. In addition to the contact thermal resistance between the two components, the effective heat transfer area required for coolant liquefaction in the first-stage condenser decreases under immersion-limit conditions, which results in heat transfer deterioration. Therefore, the immersion-limit conditions greatly influenced the heat transfer performance of the first stage of the loop thermosyphon system. The two heat transfer paths were similar from the second-stage evaporator to the heat sink, which indicates that immersion-limit conditions had little influence on the heat transfer performance of the second stage of the loop thermosyphon system.

5. CONCLUSION

The cooling process of a two-stage loop thermosyphon system was studied systematically.

(a) As the heat load was increased, the temperatures of the equipment comprising the first-stage of the loop thermosyphon system increased approximately linearly, while the temperatures of the equipment comprising the second exhibited showed little variation.

(b) When a constant temperature difference was maintained between the inlet and outlet of the heat sink, and the liquid filling ratio was 60%, both the heat source temperature and the total thermal resistance attained their lowest values under the same heat load. Therefore, the optimal liquid filling ratio for the cooling system was 60%.

(c) The heat transfer boundary of the system and two types of heat transfer limit (dry-out and immersion) were determined. The dry-out limit of the system occurred when the temperature of the evaporator vapor tube was higher than the saturation temperature and the coolant was superheated. The immersion limit occurred when the temperature of the evaporator liquid tube was lower than the saturation temperature and the coolant was subcooled.

(d) Comparisons of the heat transfer paths obtained under standard and immersion-limit operating conditions demonstrate that the immersion-limit conditions significantly influenced the heat transfer performance of the first stage of the loop thermosyphon system, but had almost no influence on the second stage, where the two sets of operating conditions produced similar results.

In conclusion, this paper provides experimental data for the industrial application of a two-stage loop thermosyphon system.

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