Transient Characteristics of a Two-Phase Thermosyphon Loop for Multichip Module

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ABSTRACT

A new thermosyphon cooling module (TSCM) has been designed, fabricated and tested to cool the multi-chip module plugged into a planar packaging system. The cooling module consists of a cold plate and an integrated condenser. With an allowable temperature rise of 56°C on the surface of the heater, the cooling module TSCM can handle a heat flux of about 2.7 $\mathrm{W/cm^2}$ using R11 as working fluid. The transient characteristics of the cooling module have been proved to be excellent: that is, when a heat load is applied inside of the system, steady state can be achieved within 10 to 15 minutes. It has been found that the length of the vapor channel between the cold plate and the condenser in addition to the ambient and the condenser temperatures affect the system performance.

NOMENCLATURE

A : heat transfer area
D : heat dissipation in MCM
h : heat transfer coefficient
P : pressure
$\dot{Q}$: heat flow rate
R : thermal resistance
R^2 : squares of residuals (in Fig. 7)
T : temperature
U : loop conductance

Subscripts

c : condenser
cc : condensation heat transfer in vapor channel
cf : forced convective heat transfer in water channel
cs : conduction heat transfer between vapor and water channel in condenser
ct : conduction heat transfer between cold plate and MCM
D : heat dissipated in MCM
dev : heat flow between system and ambience
e : evaporator (cold plate)
eb : boiling heat transfer inside cold plate
es : heat transfer between cold plate surface and liquid
ext : external
in : heat flow in
int : internal
j : heat transfer through interconnecting pipe
out : heat flow out
sat : saturation state
TP : two phase flow
w : water

I. INTRODUCTION

Proper thermal management has become very important due to increased circuit speed while ensuring high reliability of electronic circuits in communication systems. For example, switching systems in a Broadband Integrated Service Digital Network (B-ISDN) use Asynchronous Transfer Mode (ATM) and have a throughput on the order of tera-bits per second [1], [2]. With this increase in speeds, the heat flux in a B-ISDN will be one or two orders of magnitude higher than that of a conventional system, reaching 1 to 2 W/cm^2. This requires new system packaging and ingenious cooling technology. Furthermore, the next generation ISDN which will employ planar or 3-D packaging system is anticipated to have even greater heat loads and to require innovative cooling method.

The direct immersion cooling [3] has been considered as a promising methods for such application because it removes a large amount of heat effectively. However, application of such cooling method to the communication systems is not an easy task because of difficulties in maintenance and reliability. As an alternative, an ingenious air- cooled thermosyphon module [4] and an indirect liquid-cooling thermosyphon [5], [6] have been proposed for cooling high-density electronic packaging. A historical overview on thermal control of electronic components by heat pipes and two phase closed thermo- syphons was presented by Polášek and Zelko [7].

In this study, a thermosyphon cooling module (TSCM) has been designed, fab-

ricated and tested to cool the multi-chip module plugged into a planar packaging system. This cooling module is essentially a separate-type heat pipe. The cooling module consists of a cold plate and an integrated condenser. The heat from the multi-chip device is removed by thermosyphonic boiling action in a vapor channel of which width is controlled by a baffle. The vapor channels for the vapor from the cold plate and for the coolant to condense the vapor are made in one metal block. For an all0owable temperature rise of 56°C on the surface of a heater, the cooling module TSCM can handle a heat flux of about 2.7 W/cm^2. It has been found that the length of the vapor channel between the cold plate and the condenser in addition to the ambient and the condenser temperatures affect the performance of the system. Also it has been found that the module shows good transient characteristics: that is, for any given heat flux applied, the system reaches a steady state within 10–15 minutes for all cases tested. The transient characteristic which is closely related to the boiling inception inside of the cold plate is very important to such boiling-assisted cooling module design because the temperature overshoot due to retardation of boiling inception may damage the electronic chips. These results indicate that TSCM is a suitable cooling system for achieving high throughput in an ATM switching node for B-ISDN.

II. THERMOSYPHON COOLING MODULE

1. Design of Thermosyphon Cooling Module

A multi-chip module (MCM) [8] has been proposed for improving the performance of communication system. The high packaging density of an MCM will decrease signal propagation delay as well as electromagnetic emission. The proposed MCM contains a planar array of large-scale integrated chips (LSIs) as shown in Fig. 1. In this study, two 130 mm × 100 mm × 3.5 mm alu-minum-covered plate heaters were used to simulate MCM. The maximum allowable heat flow rate from this heater was about 345 W at the applied voltage of 220 V. The corresponding maximum heat flux achieved was about 2.65 W/cm^2 if one side was insulated. Heat flow rate to the insulated side was measured to be negligible. The heaters were attached to the cold plate by using OMEGA Therm-101, a thermal paste.

A schematic of TSCM is shown in Fig. 2. The cooling module consists of a cold plate and an integrated condenser. As shown in Fig. 3, the heat from the simulated MCM is absorbed in the cold plate and removed by thermosyphonic action due to boiling in a flow channel of which width is controlled by a baffle. The channel width or the baffle gap which is defined as the distance between the cold plate and the baffle may be one

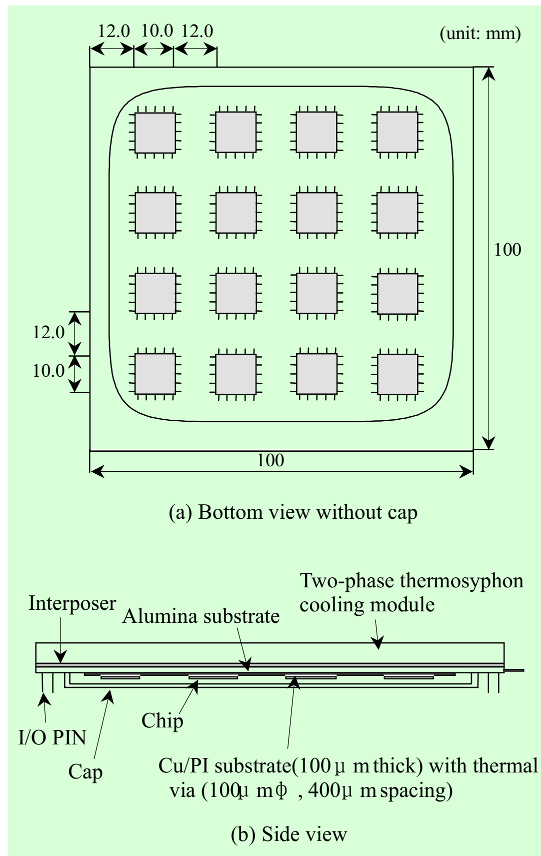

Fig. 1. Analyzed model of a multichip module [8].

of the important factors affecting thermosyphonic pumping [1], [9]. The heat transfer area for convective boiling in the cold plate is about 20 cm^2. The condenser is made of an aluminum block so that flow channels for the vapor from the cold plate and for coolant to condense the vapor are made to minimize the thermal resistance there. Eight and seven horizontal channels for vapor and liquid respectively were fabricated in the aluminum block. The channel diameters were 4 mm for the vapor and 8 mm for water with equal length of 10.3 cm per

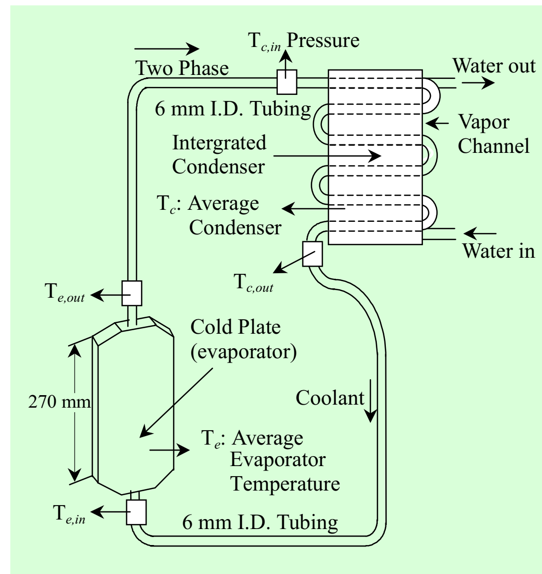

Fig. 2. Schematic of TSCM and the monitoring position of temperature and pressure.

channel. The dimension of the aluminum block is 275 mm× 100 mm× 60 mm. The center line distance between water and vapor channels is 25 mm. As can be seen in Fig. 2, the vapor from the cold plate flows in the opposite direction from the coolant in the condenser.

The coolant (water) was circulated to the condenser by a constant-temperature bath circulator. The water flow rate to the condenser, which was fixed, was about 5.58 ℓ/min. The temperature of the water at the inlet to the condenser varied depending on the preset surface temperature of the condenser. The reservoir was filled with a working fluid to the point where the baffle was submerged which included about 90% of the internal space of the cold plate. Charging of working fluid to the system is as

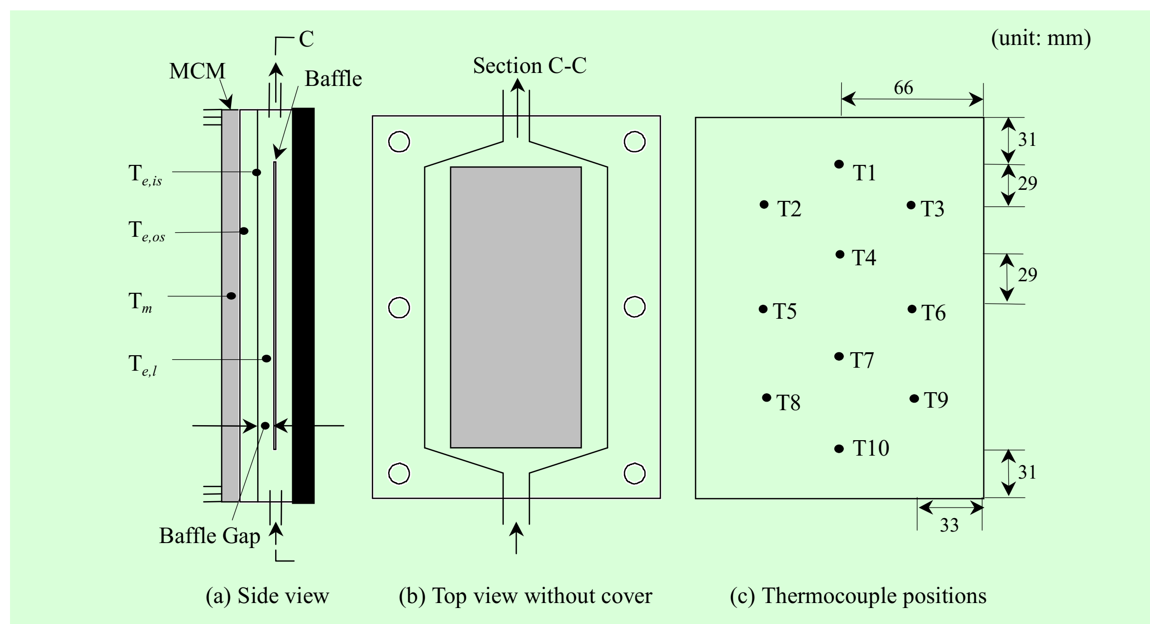

Fig. 3. Schematic of cold plate for TSCM and its thermocouple positions.

follows: first, evacuate the system up to 10 Torr by vacuum pump, and then charge the liquid to the cold plate. After the charge, the system pressure rises normally to 660 Torr. The vapor generated at the cold plate flowed to the condenser where it was condensed by the circulating water of a fixed flow rate. The liquid then returned to the evaporator by gravity. Since the vapor and returned condensation had different paths, flooding phenomena did not occur in such a separated-type closed thermosyphon unit [8] to allow more heat to be transported.

The temperatures of the cold plate and the condenser were measured using twenty T-type thermocouples mounted on the surfaces as shown in Fig. 3. Acquisition of data obtained from the T-type thermocouples was done by a Yokogawa recorder (HR-1300) connected to a PC. The data collected from each thermocouple at two second intervals over two minutes were averaged separately to make a data set. Also the vapor temperatures at the inlet of condenser $T_{c,in}$ and at the outlet of the cold plate $T_{e,out}$ and the liquid temperatures at the inlet of the cold plate $T_{e,in}$ and at the outlet of the condenser $T_{c,out}$ were measured by T-type thermocouples whose monitoring positions are shown in Fig. 2. Acquisition of data obtained from these thermocouples was done by a Yokogawa recorder (LR-8100). The uncertainties in the temperature measurements are less than ± 0.1°C. The system pressure was measured using a piezoelectric pressure transducer (OMEGA).

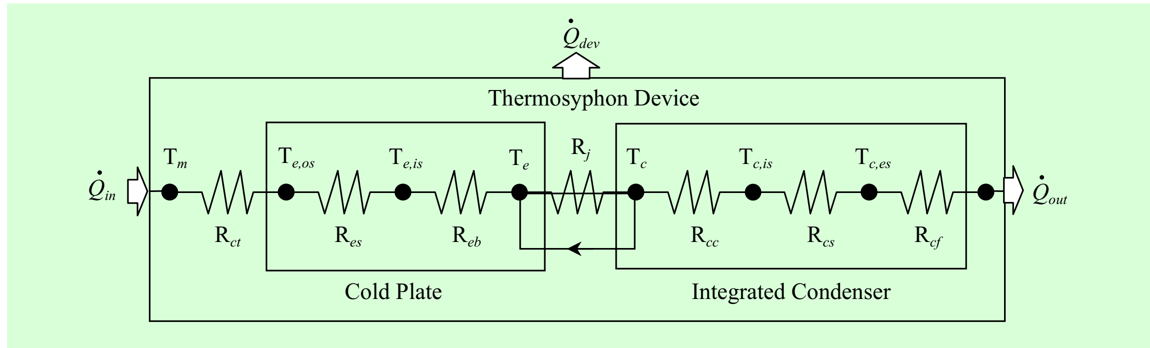

Fig. 4. Overall heat balance and heat removal process in TSCM.

2. Principle of Operation of a Thermosyphon Device

The first law of thermodynamics for any thermosyphon device at a steady state may be written as

$$\dot{Q}_{in} + \dot{Q}_{dev} + \dot{Q}_{out} = 0, \tag{44}$$

where $\dot{Q}_{in}$ and $\dot{Q}_{out}$ are the heat flow rates to and from the system, respectively, and $\dot{Q}_{dev}$ is the heat transfer rate to environment through the vapor channels and the surfaces of cold plate and condenser. Effective absorption of the heat input $\dot{Q}_{in}$, rapid transport of the absorbed and adequate heat removable to environment $\dot{Q}_{out}$ are all important factors for smooth operation of the device.

A schematic of the overall heat balance and the heat removal process of TSCM at a steady state are shown in Fig. 4. The heat generated in MCM $\dot{Q}_{in}$ is conducted to a coolant channel through the medium of an interposer and the cold plate wall having the thermal resistances of R_{ct} and R_{es}, respectively. The temperature at the surface of MCM T_m and the average temperature of the cold plate wall $T_e (T_{e,os} < T_e < T_{e,is}$, where $T_{e,os}$ and $T_{e,is}$ are the temperatures at the inner and outer wall of the cold plate as shown in Fig. 3) are mainly determined by capability of the heat absorption in the cold plate. The amount of heat absorption is characterized by the thermal resistance related to boiling action in the channel, R_{eb}. The vapor temperature in the condenser is solely determined by the system pressure which is governed by the overall heat transfer process of the system. The transport resistance R_j due to heat and pressure losses in the vapor interconnecting pipe turned out to be small [10]. The resistance may be defined as

$$R_j = [T_{sat}(P_e) - T_{sat}(P_c)]/\dot{Q}_{in}. \tag{45}$$

On the other hand, the liquid temperature $T_{e,l}$ at the bottom of the cold plate is affected by the degree of sub-cooling of the condensate. The flow rate and the inlet temperature of water to the condenser $T_{w,i}$, which control the average temperature of

the wall between vapor and water channels T_c ($T_{c,i} < T_c < T_{c,e}$, where $T_{c,i}$ and $T_{c,e}$ are the temperatures at the wall of the vapor and water channels inside the condenser, respectively.) affect the temperature of the condensate as well.

Table 1. Thermal resistances (internal) and effective areas of heat transfer at each junction for TSCM.

	Thermal resistance (K/W)	Effective heat transfer area (cm^2)
R_{ct}	0.0002	A_{ct} = 26.0
R_{es}	0.001	A_{es} = 26.0
R_{eb}	0.020	A_{eb} = 20.0
R_j	···	···
R_{cc}	0.060	A_{cc} = 15.0
R_{cs}	0.002	A_{cs} = 26.0
R_{cf}	0.010	A_{cf} = 25.0

In TSCM, the resistances related to the conduction heat transfer such as R_{ct}, R_{es} and R_{cs} which are determined by the physical dimensions and the thermal conductivity of materials are smaller in order of magnitude scale than the thermal resistances of R_{eb}, R_{cc} and the forced convection in the water channel R_{cf}. The thermal resistances and the effective areas of the heat transfer at the heat flux of 2.22 W/cm^2 with a baffle gap of 10 mm at room temperature of 23–25°C are shown in Table 1. The areas denoted by A_{ct}, A_{es} and A_{cs} in Table 1 are contact areas and the other ones are the effective heat transfer areas. The thermal resistance values shown in this Table suggest that proper handling of condensation process is one of crucial factors for consideration of the thermosyphon module design.

During steady state operation where estimation of the system's performance is possible, the thermal resistance of boiling inside the cold plate is given by

$$R_{eb} = \frac{1}{h_{TP} A_{eb}}, \tag{46}$$

where the heat transfer coefficient for two phase flow h_{TP} may be obtained by Chen's correlation [11] and A_{eb} is the effective heat transfer area for boiling. Carey *et al.* [5] was successful to predict the heat transfer coefficient from such cold plate type with offset strip fins by using Chen's correlation. One may obtain overall heat transfer coefficient in the cold plate by employing the correlation developed by Imura [12]. However this correlation may not be used to predict the local heat transfer coefficient which depends on void fraction and quality. The thermal resistances related to the condensation of vapor and the forced convection of water may be obtained by using existing correlations [13] for appropriate heat transfer areas. The evaporation resistance is increased whenever a part of the cold plate surface ceases to be in contact with a two-phase mixture. This may occur at the bottom of the cold plate where sub-cooled liquid enters or at the top where drying-out occurs. The condensation resistance may be increased whenever the effective area for condensation is reduced as a result of increase in liquid film thickness at the bottom of the condenser.

System's performance at a steady state may be evaluated by the loop conductance U [11] or the corresponding external thermal resistance [4], which are defined as

$$U = \frac{\dot{Q}_{in}}{A_e(T_e - T_c)} \tag{47}$$

and

$$R_{ext} = \frac{T_e - T_c}{\dot{Q}_{in}}. \tag{48}$$

The above equations may provide criteria to compare the performance of various thermosyphon devices. Also the chip junction temperature may be written as follows [4]

$$T_{c,m} \simeq T_c + R_{int} D < 85°C, \tag{49}$$

where D is the amount of heat dissipated in MCM and $T_{c,m}$ is the device junction temperature. With the internal resistances estimated shown in Table 1, TSCM can handle power as much as 730 W at a steady state with $T_c = 15°C$. Different values for the loop conductance and the external thermal resistance may be obtained if one use the inlet temperature of coolant instead of T_c [4].

III. RESULTS AND DISCUSSIONS

Experiments for the designed cooling module were performed without insulation of a condensing part, condenser wall, or tube connecting the condenser with an evaporator, in order to simulate more accurately the real conditions of electronic chip cooling. Experiments were performed to find out how the thermal performance of the designed cooling module is affected depending on several parameters such as ambient temperature, presence of a baffle, and the tube length between an evaporator and a condenser. A great emphasis was put on the transient characteristics of the cooling module, which has never been studied previously. In fact, for the transient state where the heat balance given in (1) is not valid, an appropriate prediction method is not available at present. At present, a proper method is not available to describe the transient characteristics of two-phase thermosyphon cooling device, which is closely related to the boiling inception inside of the cold plate. Usually the cold

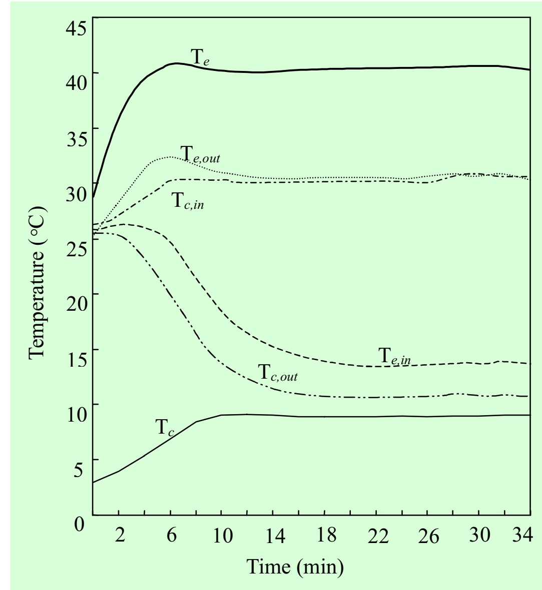

Fig. 5. Transient characteristics for TSCM with heat flux of 1.08 W/cm^2, R11 as coolant and baffle width of 5 mm.

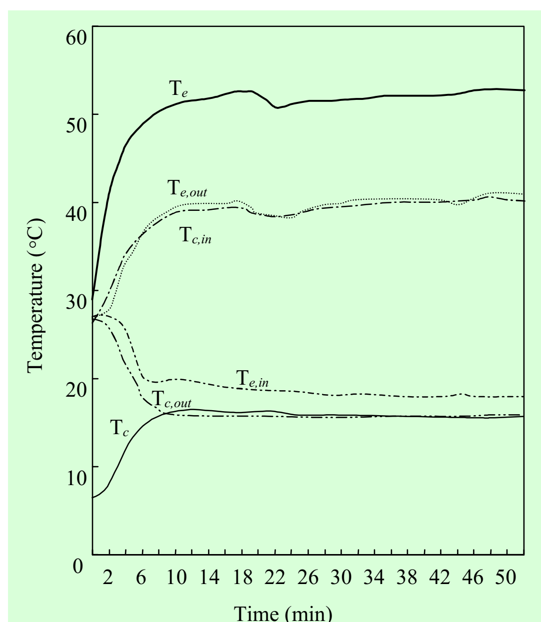

Fig. 6. Transient characteristics for TSCM with heat flux of 2.22 W/cm^2, R11 as coolant and baffle width of 5 mm.

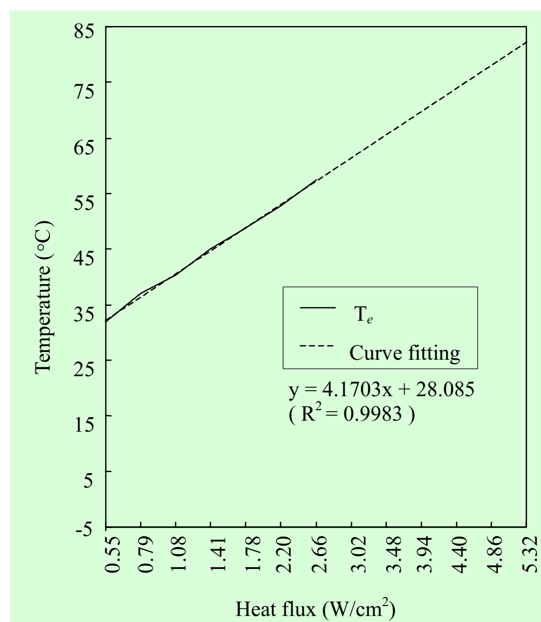

Fig. 7. Average temperature of the cold plate surface at steady state depending on heat flux applied R11 as coolant and baffle width of 5 mm.

plate design for the thermosyphon cooling device has to be done under the assumption of the steady state condition [5], [14]. Further, the transient characteristics of TSCM was tested to the case of the time-dependent heat flux. The cooling module TSCM responded remarkably to the time dependent, step-wise heat flux variation. The test was done at various heat fluxes of 0.55 W/cm^2, 0.79 W/cm^2, 1.08 W/cm^2, 1.41 W/cm^2, 1.78 W/cm^2, 2.22 W/cm^2 and 2.66 W/cm^2. It took one hour or so to complete each test. Also the thermodynamic performance of tested TSCM was compared with the other thermosyphon devices. Some of the results are as in the following sections.

1. The Effect of Varying Ambient Temperature with Uniform Heat Flux

A. Case of 5 mm Baffle Gap with an Uncontrolled Ambient Temperature

In Figs. 5 and 6, respectively, test results for heat fluxes of 1.08 W/cm^2 and 2.22 W/cm^2 are presented for the case of R11 as a coolant, ambient temperature of up to 30°C and a baffle gap of 5 mm. It is shown by these results that the temperature of the simulated chip reaches a steady state within 10 minutes and there is no heat transfer crisis. It is shown that the average temperature rise of the simulated chip is just 12°C though heat flux is doubled. One may expect from the result in Fig. 7

that TSCM can cool electronic chips of which heat dissipation is up to 5.0 W/cm^2 maintaining its temperature at below 75°C. Since the maximum heat flux load of immersion cooling technology using R11 is less than 10.0 W/cm^2, TSCM is considered as an attractive cooling module. It has also been proved that TSCM can handle time-dependent heat loads.

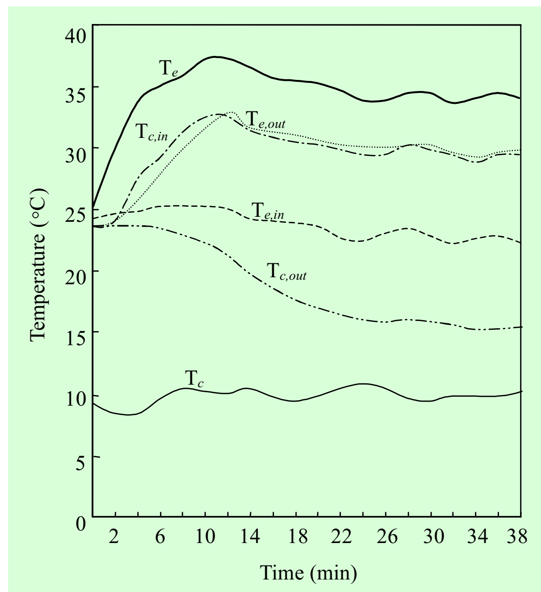

Fig. 8. Transient characteristics for TSCM with heat flux of 1.08 W/cm^2, baffle width of 5 mm and R11 as coolant at room temperature of 23–25°C.

B. Case of 5 mm Baffle Gap with a Controlled Ambient Temperature

Test results are presented for the case of R11 as a coolant, maintaining ambient temperature at 23–25°C and a baffle gap of 5 mm when heat fluxes of 1.08 W/cm^2 and 2.22 W/cm^2 were applied. The results are

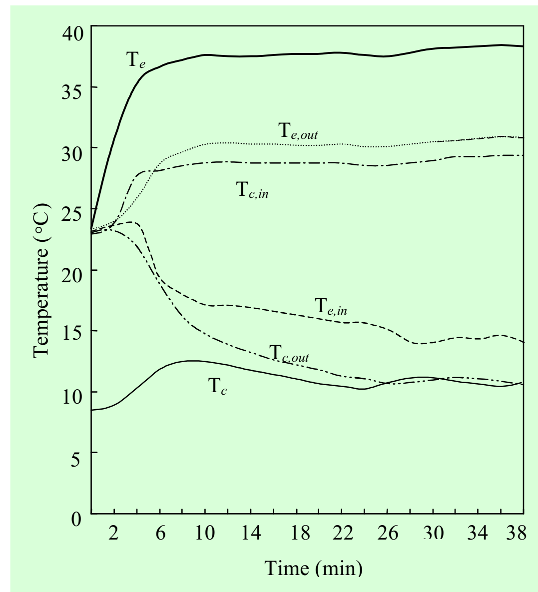

Fig. 9. Transient characteristics for TSCM with heat flux of 2.22 W/cm^2, baffle width of 5 mm and R11 as coolant at room temperature of 23–25°C.

shown in Figs. 8 and 9, respectively. The reason why such baffle gap size was chosen is that similar boiling characteristics have been obtained with variation of the baffle gap between 1 mm and 4 mm for the submerged vertical channel in FC-72 [6]. The heat transfer crisis appears distinctly at a low-heat flux, as compared to the result of Fig. 5 at the same heat flux for the case A. However, as the heat flux increases, the phenomena of heat transfer crisis disappears gradually and the steady state surface temperature of a simulated chip for the case of a controlled ambient temperature appears to be lower than that for the case of an uncontrolled ambient temperature. When the heat flux is 2.22 W/cm^2, the average temperature of an evaporator at a steady state

(T_e) is shown to be lowered by 14°C compared to the case of the uncontrolled ambient temperature as shown in Fig. 6, which is still a significant result since the environmental temperature is lowered by only 7°C. A bit of fluctuation in the temperatures of the cold plate and condenser surfaces occurred for the case shown in Fig. 8. Such mild oscillation might be due to slight variation of the temperature of the condensate [15], which, in turn, caused the fluctuations in boiling inception.

2. Results Depending on the Variation of the Baffle Gap

As found from the test results showing that the constant ambient temperature is favorable for cooling at the high heat flux, the ambient temperature was held at 23–25°C for testing baffle gap variation.

The test results for a baffle gap of 10 mm and that of 5 mm were compared to investigate the cooling performance depending on variation of a baffle gap. The average temperature of the simulated chip surface was almost the same when it reached a steady state. However, for the case of a baffle gap 10 mm, the phenomena of heat transfer crisis decreased remarkably for the lower heat flux of 1.08 W/cm^2, and showed more thermal stability at a steady state for the higher heat fluxes applied.

Under the condition of a controlled ambient temperature with 10 mm baffle gap, the surface temperatures on the cold plate and condenser at each position shown in Fig. 4 were measured to investigate the detailed transient characteristics of the cooling module. The results for the heat fluxes of 1.08 W/cm^2 and 2.22 W/cm^2 are shown in Figs, 10 and 11, respectively. As can be seen in Fig. 10(b), the characteristic temperature of the evaporator is primarily determined by the thermal response of the lower part, where the temperature is much higher than that of the upper position. This is due to the fact that boiling action is hardly achievable at the lower part of the cold plate. In the condenser, the temperature of the upper part where vapor from the cold plate flows is higher than that of lower part where condensate circulates as shown in Fig. 10(c). The thermocouple positions imbedded in the condenser surface is very similar to the ones in the cold plate shown in Fig. 3. The first thermocouple is located at the distance of 30 mm from the top and the second and third ones are located at the distance of 65 mm from the top with separation of 40 mm each other. Thus the phase change occurring on the cold plate as well as in the condenser turned out to be typically non-equilibrium. As seen in Figs.10(b) and 10(c), the ambient temperature provides a demarcation line on the average surface temperatures of the evaporation and the condenser.

3. Results with Fixed Condenser Temperature and Variation of Tube Length between Cold Plate and Condenser

i) The cooling performance test for the case of setting the temperature of condenser

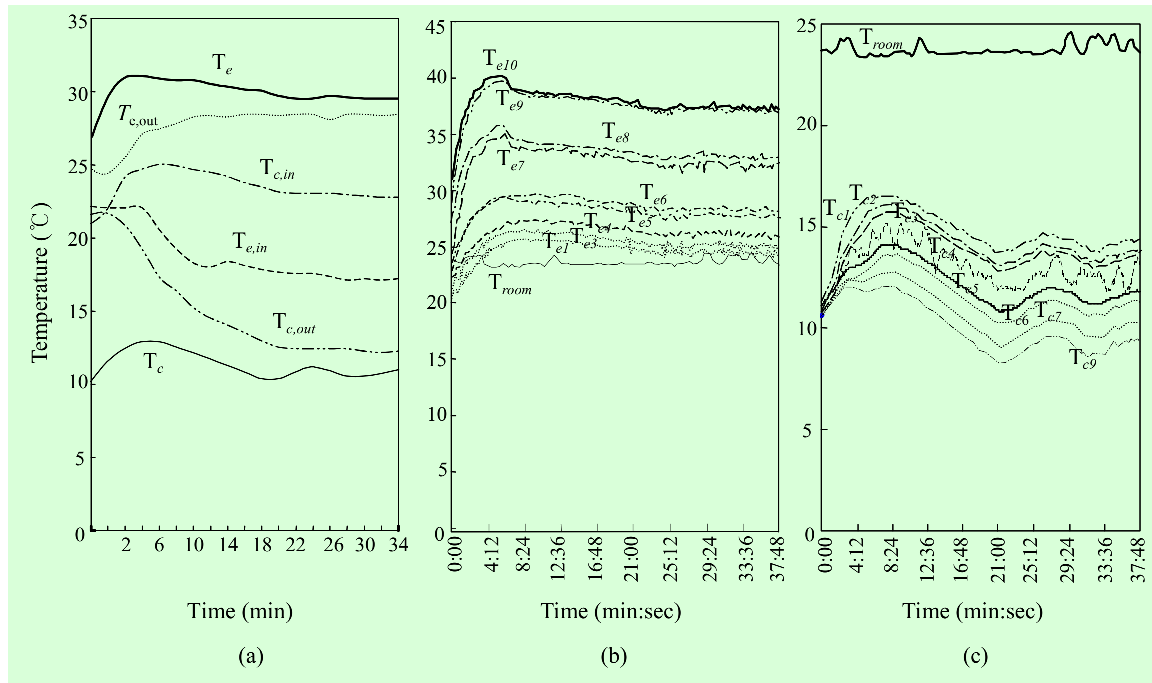

Fig. 10. (a) Transient characteristics for TSCM with heat flux of 1.08 W/cm^2, baffle width of 10 mm and R11 as coolant at room temp. of 23–25 °C, (b) Temperature trend on TSCM cold plate with heat flux of 1.08 W/cm^2, baffle width of 10 mm and R11 as coolant at room temp. of 23–25 °C, (c) Temperature trend on TSCM condenser section with heat flux of 1.08 W/cm^2, baffle width of 10 mm and R11 as coolant at room temperature of 23–25 °C.

surface at 11.5°C was carried out. The reason for this temperature setting was to have possible enhancement of the cooling performance due to acceleration of evaporation and condensation. However the test results were not consistent with this expectation. The temperature overshoot appeared notably and the temperature at a steady state was considerably high, which could affect unfavorably the cooling performance. This indicated that the best performance of the module might be achieved when the cooling capacity of the cold plate was balanced with that of the condenser. Excess cooling at the condenser lowered the temperature of the condensate, which induced difficulty in boiling inception inside of the cold plate. As seen in Fig. 12, the vapor pressure in the system followed the trend of the average temperature of the evaporator as expected.

ii) Another test for the case of the length of 60 cm between the evaporator and condenser instead of 100 cm was performed to examine how the height affected the cooling performance. The results showed that the phenomena of heat transfer crisis appeared to be remarkably increased and the temperature at a steady state was considerably

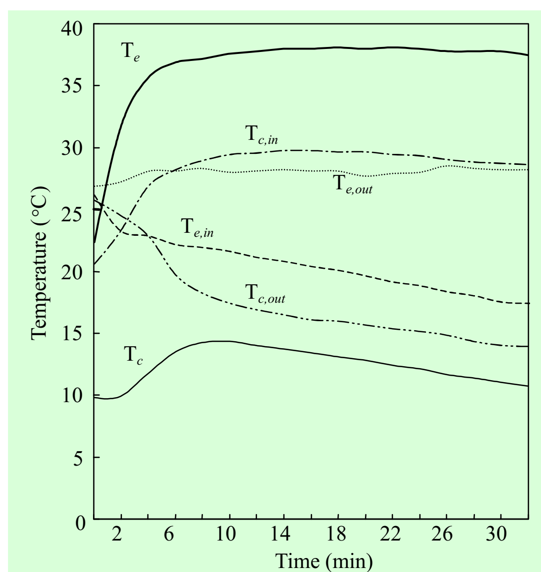

Fig. 11. Transient characteristics for TSCM with heat flux of 2.22 W/cm^2, baffle width of 10 mm and R11 as coolant at room temp. of 23–25 °C.

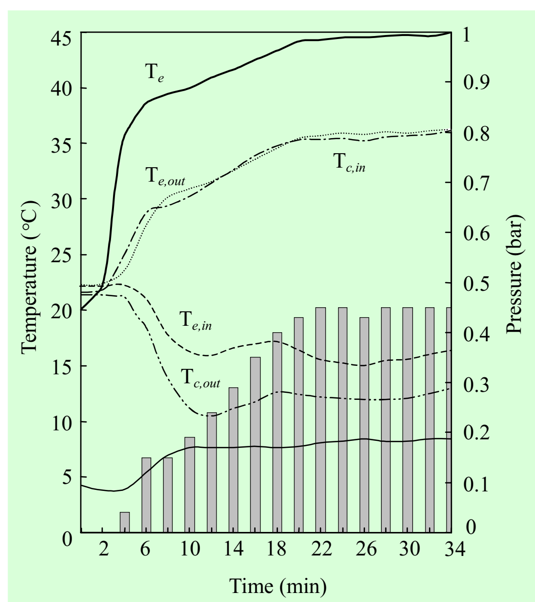

Fig. 12. Transient characteristics for TSCM with heat flux of 2.22 W/cm^2, baffle width of 10 mm and R11 as coolant at room temperature of 24–27°C and condenser surface temperature of 11.5°C.

high, which could affect unfavorably the cooling performance, which is clear when comparing Fig. 13 with Fig. 10(a). Therefore, it may be supposed that, in the case of the height of 60 cm compared to the height of 100 cm, the change in temperature of the condensing part and condensed coolant moving down from the condenser be made very slowly. This may be due to the fact that if the length of the vapor interconnecting pipe is smaller, the vapor pressure in the condenser becomes higher.

Finally, it was noted that the cooling module developed did not work with a working fluid of FC-72 under the conditions tested. This may be due to a relatively higher boiling point of FC-72. It is better to use a working fluid of which boiling point is close to or lower than the ambient temperature for cooling of electronic devices by using this module.

IV. DESIGN RECOMMENDATIONS

The experiments mentioned above show that the heat transfer crisis is the most important factor affecting transient characteristics of the thermosyphon cooling module. It has been found that factors related to the heat transfer crisis of the cooling module include

ambient temperature;

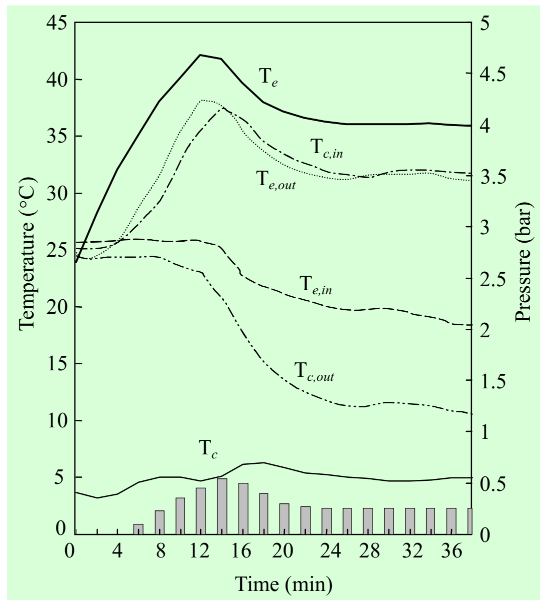

Fig. 13. Transient characteristics for TSCM with heat flux of 1.08 W/cm^2, baffle width of 10 mm and R11 as coolant at room temperature of 23–26 °C, the tube length between evaporator and condenser of 60 cm.

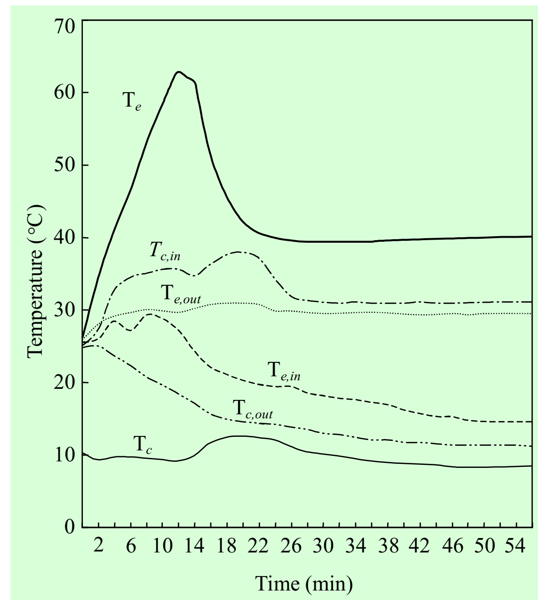

Fig. 14. Transient characteristics for ITSM with heat flux of 1.08 W/cm^2 and R11 with 40% loading as coolant in the evaporator section.

width of baffle gap;
surface temperature of the condenser;
and, tube length between the condenser and the evaporator.

To achieve the maximum thermal performance, the external thermal resistance should be minimized. The external thermal resistances for the module tested, which are defined in (5), are in the range between 0.06 K/W and 0.12 K/W. These values are considerably lower than those of NTT's (Nippon Telegraph and Telephone Corporation) cooling module. It is found that the thermal resistance is lower at higher heat input and is lower than the summation of all internal thermal resistances of the system R_{int}, that is about 0.09 K/W for the heat flux of 2.22 W/cm^2 with a baffle width of 10 mm and R11 as the coolant at room temperature of 23–25°C. NTT's cooling module with the integrated condenser (ITSM) employed in this study shows significant heat transfer crisis at the lower heat flux and considerable temperature rise of an evaporator at the higher heat flux as shown in Figs. 14 and 15, which suggests some difficulty in boiling inception on the surface of the liquid-filled horizontal tube. Eight horizontal channels where evaporation of working fluid takes place were made inside a metal block, NTT's cold plate. To be noted is that in the NTT's cooling module with air-cooled condenser the average

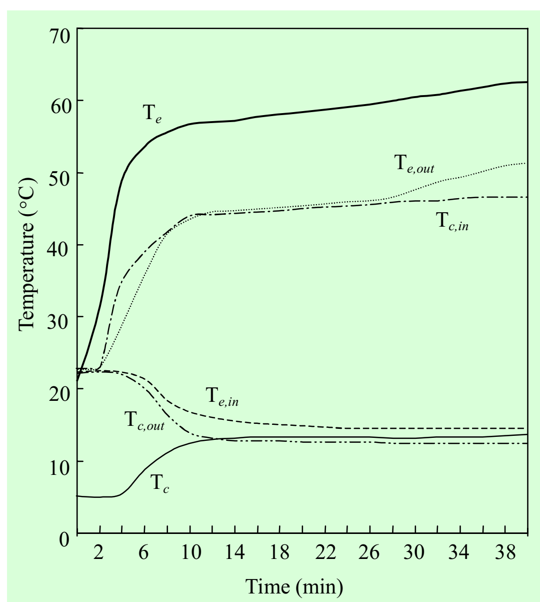

Fig. 15. Transient characteristics for ITSM with heat flux of 2.22 W/cm^2 and R11 with 40% loading as coolant in the evaporator section.

external resistance is about 0.2, which is about two times larger than that of TSCM. Also the loop conductance of TSCM which is about 830 W/m^{2}K at the heat flux of 2.2 W/cm^2 is comparable to the maximum conductance of 1000 W/m^{2}K obtained for a thermosyphon device which is consisted of a rectangular loop of copper tube [9].

Based on the results and discussions in this study, the design factors for the thermo-syphons type of cooling module are to minimize the heat transfer crisis at the lower heat fluxes, to improve the thermal performance for condensation process, to decide the optimal height between the evaporator and condenser, and to determine the optimal width of the baffle gap.

V. CONCLUSIONS

A two-phase thermosyphon cooling module was developed for MCM to be used for B-ISDN. The cooling module consists of a cold plate and integrated condenser. Test results revealed that the module had good transient characteristics to the heat load applied and lower external thermal resistance, which was one of the attractive features of this module. The cooling module developed enabled heat flux levels to lie in the realm of indirect liquid cooling. It was found that TSCM developed was a suitable cooling system to achieve high throughput in an ATM switching node for B-ISDN. Furthermore, this cooling module was easily manufactured and maintained, and no signs of instabilities related to two-phase flow were found during the operation.

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