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A Study on Development of Liquid Cooled Plate for Cooling of a Communication Electronic Device with High Heat Generation

고발열 통신용 전자부품 냉각을 위한 고성능 수냉식 냉각판 개발에 관한 연구

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Key Words : Amplifier, Cooling plate, Liquid cooling system, Local heat flux, Telecommunication equipment

요약 : 통신용 전자기기에서 대부분의 열은 증폭기에서 발생한다. 일반적으로 증폭기를 냉각하기 위하여 공랭식을 사용하여 발생하는 많은 열을 냉각하였다. 그러나 전통적인 방법은 고성능 콤팩트화 되어가는 추세에서 발열되는 열을 충분히 냉각하기는 부족하다. 본 논문은 고발열 전자부품 냉각을 위해서 수냉식 방법을 사용하였다. 열전달 효율을 높이기 위하여 냉각판에 직접 냉각수를 흐르게하여 접촉저항을 줄였다. 그리고 냉각판의 유로에 대한 배열과 유량의 비의 효과를 조사하였다. 연구를 수행한 결과, 다음과 같은 결론을 얻을 수 있었다. 냉각수 순환량이 3 *l*/min인 경우, 유로 직경이 8 mm일 때의 냉각 성능이 10 mm일 때보다 우수한 것으로 나타났다. 냉각수 순환량이 3 *l*/min인 경우, 유로 직경이 8 mm일 때의 발열 소자 표면 온도 분포가 더욱 안정적으로 나타났으며, 상하부에 설치된 발열 소자 표면 온도가 더 낮게 나타났다. 동일한 유로 직경의 냉각판에서, 열유속 증가에 따른 냉각수의 전열 성능 증가로 인해 전체 발열량의 증가율보다 발열 소자의 온도 증가율이 낮게 나타났다.

Nomenclature

m : flow rate of a cooling water [kg/s]
Cp : specific heat [J/kg·K]
Q : heat capacity [W]
T : temperature [℃]

Subscripts

c : calculated value
h : input heat quantity of heat
i, o : input, output
s : surface
w : cooling water

1. Introduction

Among components used in telecommunication equipment an amplifier generates most amount of heat. This makes the device temperature rise. The large temperature rise produces thermal stress in the equipment, and makes the life system of electronic components short. In general, an amplifier can be categorized as High Power Amplifier (HPA) and Linear Power Amplifier (LPA). HPA has a small output, and it generates a small amount of heat. It is usually used in a small sized telecommunication module. A cooling method of HPA has an air-cooled type. LPA has a large output. It shows low performance efficiency and generates a lot of heat. If LPA has an air-cooled cooling type, it causes a large temperature gradient. That is undesirable in the system. Also it causes a lot of structural and economical loss. Kim et al.(1) investigated a numerical simulation on cooling plates in a fuel cell. In this paper, they suggested the best performance configuration of the proposed cooling plated models. Son and Shin(2) studied a numerical analysis on cooling characteristics of electronic components using convection and conduction heat transfer. They used the five different cooling

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methods to find efficient cooling method in a given geometry and heat source. Kim et al.(3) researched about a liquid cooling plate with a constant heat flux condition. A heat flux condition is 0.685 W/cm^2, and ambient temperature is 26℃. Experimental results of the cooling plate show 37% increased performance compared to an air-cooled type. Recently, telecommunication systems have led to the significant increase on power densities in equipments. To get a stable system performance, a reliable cooling system must be considered. Cooling method using liquid coolant(4,5), which is one type of heat transfer, in a telecommunication module has actively researched. Previously, electronic components of a chip unit are conducted with direct cooling types. It is obvious that successful design of a telecommunication module based on liquid cooling system requires fundamental understanding. In this paper a cooling method of an indirect cooling type in a cooling plate of a module unit is investigated. A selected model which has locally heated condition, without a constant heat flux condition, is studied. Results can be used to increase performance of an amplifier module.

2. Experimental apparatus and procedure

Fig. 1 shows a diagram of experimental apparatus and flow loop. The experimental facility consists of a circulation pump, copper piping, coolant tank, cooling fan, and test section. By a circulation pump the working fluid is pumped from the liquid tank. The working fluid absorbs heat which is generated at module components, and flows continuously to a cooling section. The temperature of the working fluid is monitored, at the entrance and exit of the test section. At the radiant section the working fluid's temperature becomes constant and then recalculates. The cooling section has an air-cooled type cooling system. By controlling fan velocities, the temperature of the working fluid can be cooled to the specified temperature.

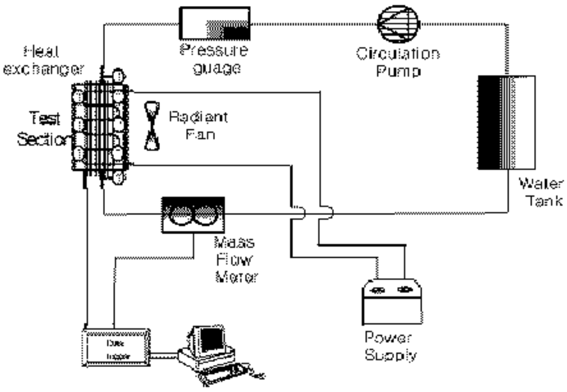

Fig. 1 Experimental apparatus.

Fig. 2 shows a schematic diagram of heaters in the amplifier module. It shows the geometry of flow passage in a cold plate. To increase cooling efficiency, heat components are located above and below the flow passage, respectively. This means that heat generating components installed at the front side are also located at the back side. The test plate is made by an aluminum plate. This material has a high thermal conductivity and is economical.

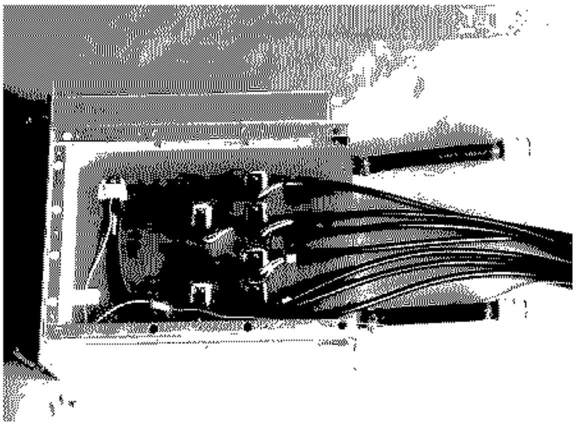

Fig. 2 Photo of the experimental amplifier module and heaters.

Table 1 Experimental conditions.

Classification		Unit	Value	
Temperature	Test room	K	300	
	Cooling water		290	
Flow rate	Cooling water	ℓ/min	3	
Heat Transfer rate(Fig.3)	① ~ ④	W/m^2	59	79
	⑤ ~ ⑥		20	26
	⑦		28	37
Flow passage diameter	Cooling plate	mm	8, 10	

The circulation pump is activated to pump the working fluid. Circulation quantity of the coolant is measured by a flow meter, which is controlled by a valve. Then electrical power on the heating devices is adjusted to have the desired temperature level. All the heat generating components of the module are individually checked by a control panel which can be set to generate the specific quantity. That heat quantity can be checked by a display panel. The temperature of the cooling liquid becomes high

after passing the whole heating components. The cooling liquid by a controlling the radiant fan can have the specified temperature, which is ready to re-circulate. Temperature is measured at the inlet and outlet in the tank, and radiant components. To collect temperature profiles, a data log (Yokogawa, MX 100) is used.

In general temperature range of a telecommunication cooling system varies between -30°C ~50°C. This means a system must have a stable operation within this temperature range. For a proper operation of the system, a cooling liquid is the most important factor. An anti-freezing solution as a cooling liquid is added to cooling water not to freeze at -30°C. Next, radiant components must maintain at acceptable temperature levels which are below 95°C. To achieve this desirable condition, flow passage geometry, flow rates of a cooling liquid, and generated heat quantity are considered. The geometry of a flow passage has two types after considering performance and manufacturing factors. And the cooling quantities of a module, after considering output range, have two conditions of 600 W/m^2 and 800 W/m^2. Table 1 shows experimental conditions.

Heat loss to the ambient was estimated at less than 1% of the total heat input. Therefore, heat losses were assumed negligible, and cartridge heater measured by the wattmeter was used for all heat flux calculations. Excellent agreement between electrical power input and measured enthalpy change of the cooling water further verified this assumption. Error associated with the thermocouple measurements was estimated at less than 0.3 °C.

3. Mathematical model

In this section, the three-dimensional fluid flow and heat transfer characteristics of the cooling plate are first analyzed numerically. Then, the measured performance presented and compared with the numerical predictions.

Heat transfer in the unit cell is a conjugate heat transfer which is the heat conduction in the solid and the convection to the cooling fluid. The two heat transfer modes are coupled by continuities of temperature and heat flux at the interface between the solid and fluid$^{(6)}$, which are expressed as

$$T_{s,\Gamma} = T_{f,\Gamma} \tag{1}$$

$$-K_s \frac{\partial T_s}{\partial n}\Big|_\Gamma = -K_f \frac{\partial T_f}{\partial n}\Big|_\Gamma \tag{2}$$

Several simplifying assumptions are incorporated before establishing governing equations for the fluid flow and heat transfer in the unit cell:

(1) steady fluid flow and heat transfer;

(2) incompressible fluid;

(3) laminar flow;

(4) negligible radiation heat transfer;

(5) constant solid and fluid properties except water viscosity;

(6) negligible natural convection of air

Further explanations are needed for assumption (5). In assumption (5), the solid and liquid properties are assumed constant because variations of these properties are small within the temperature range tested. However, the variation of water viscosity is significant. Therefore, variable temperature dependent water viscosity based on mean temperature is employed.

Based on the above assumptions, the governing differential equations used to describe the fluid flow and heat transfer in the unit cell are expressed as follows. For the cooling water, the continuity, momentum, and energy equations are expressed, respectively, as

$$\nabla \vec{V} = 0, \tag{3}$$

$$\rho_f(\vec{V}\cdot\nabla\vec{V}) = -\nabla P+\nabla\cdot(\mu_f\nabla\vec{V}), \tag{4}$$

$$\rho_f c_{p,f}(\vec{V}\cdot\nabla T) = k_f\nabla^2 T. \tag{5}$$

For the solid regions, the continuity and momentum equations are simply

$$\vec{V} = 0, \tag{6}$$

and the energy equation is

$$k_s\nabla^2 T = 0. \tag{7}$$

A numerical method to solve this conjugate heat transfer problem is to treat the solid and fluid as a unitary computational domain and solve the above governing equations simultaneously.

Fig. 3 illustrates the scheme of the test section for numerical analysis. The grid system employed in the present numerical analysis has 102, 22, and 43 nodes in the x, y and z directions, respectively. A non-uniform grid arrangement in the x-direction, with a larger number of grid points near the channel inlet, is used to resolve the developing region. Dimensions of the cooling plate are given in Table 2.

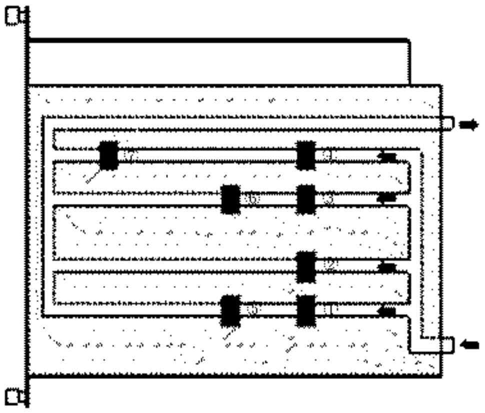

Fig. 3 Schematic diagram of the cooling plate.

Table 2 Dimensions of the cooling plate.

Cooling plate	Length [mm]	300
	Width [mm]	280
	Height [mm]	30
Channel	Diameter [mm]	10
Module	Dimension [mm×mm]	30×25

4. Results and discussion

Fig. 4 shows the results of heat balance of the module in order to check the experiment accuracy. Electronic components attached on the cold plate in the experimental amplifier module do not have an adiabatic condition, but a convection boundary condition. This means that the surface is exposed to an ambient air. Heat loss to the ambient can be measured by the recovery heat amount by the cooling liquid. The quantity Q is calculated as

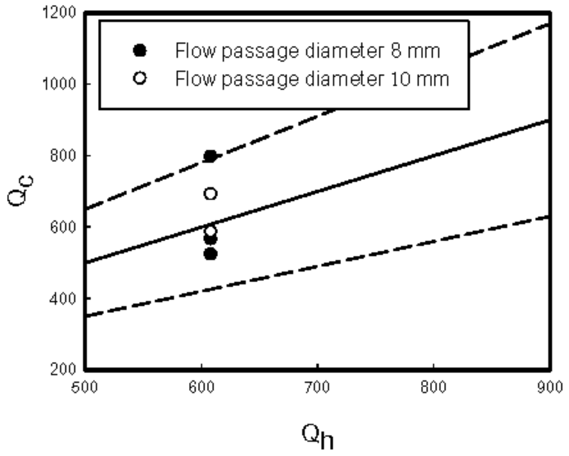

Fig. 4 Heat balance for the cooling plate.

$$Q = m\times C_p\times(T_o - T_i) \tag{8}$$

Calculated results based on experimental results show the error ranges of -14%~+30% compared to actual input of electric power. To make the installation and modification of the cooling plate easy, a rubber hose and a couple ring are used. This causes relatively a large error because at low flow rate flow resistance occurred at the couple ring is large. When flow rate is large which is the case of a flow passage diameter is

10 mm, the results show a good balance.

A comparison between measured and predicted the surface temperature profiles with the different diameters of the channel at each module position is shown in Fig. 5. It shows good agreement between the measured and predicted surface temperatures. At this condition heat quantity is 600 W/m^2. Both cooling plates with different diameters have the temperatures below about 85 °C. This means that cooling performance is satisfactory. When a flow passage diameter is 8 mm at the given flow rate, the temperature of radiant components is lower than that of pipe diameter 10 mm. However, overall temperature tendency shows a stable profile for both cases. This result is expected when the heat balance experiment is conducted. When the flow rate is small, the inner surface of a pipe can not contact fully. When the diameter is 8 mm, the flow speed is relatively large compared to the 10 mm diameter case. This result shows good heat transfer performance. The temperature profiles of radiant components ①~④, which have the same heating capacity, are compared. When the diameter is 8 mm, the temperature distribution shows a stable status. This is because the pressure of the cooling liquid at the 8 mm diameter pipe uniformly affects the inner surface of the pipe. In case of the 10 mm pipe diameter, the surface temperature of radiant components at the middle positions shows higher than that at very end positions. This means the flow rates at the middle layer are low, because the pressure of the cooling liquid is low. At the middle layer near a tangential direction affected by static pressure the flow rates are smaller than those of mainstream of up and down directions affected by dynamic pressure. To prevent this unbalanced cooling phenomenon, the flow rates are increased. Thus, each flow passage will have equal amount of the cooling liquid.

Fig. 6 shows the surface temperatures of heaters with varying heating rates at different positions. Under the worst condition, operated at the maximum performance of a module, the surface temperature is below 95°C. This result of a temperature profile in the cold plate is satisfactory. In case of the maximum performance, the radiant heat quantity increases about 33%. However, the local surface temperature of heaters also increases from 4% to 11%. When the heat flux increased, the increase of the temperature difference between radiant components and the cooling liquid make a good performance of an anti-freezing solution. It is because that the loss to an ambient by natural convection increases. Local heat quantity of components ①~④ has about two times larger than that of components ⑤~⑦. The increase rate of the temperature also shows two times lager as heat quantity increases. The increase of the

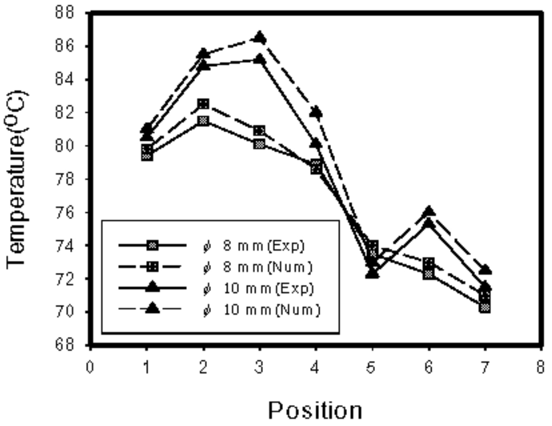

Fig. 5 Comparison of measured predicted values of surface temperature.

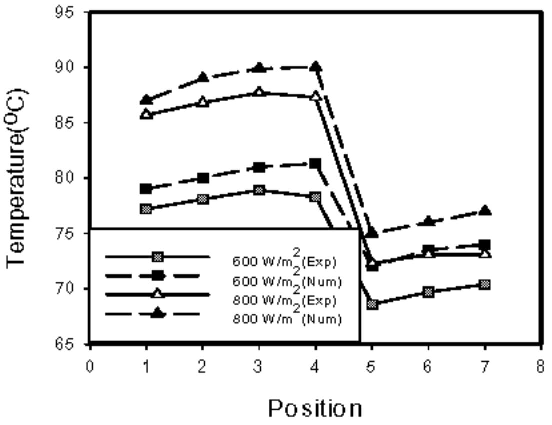

Fig. 6 Surface temperature with respect to the different supplied power.

temperature between radiant heaters shows a linear profile. This means that as the difference of heat quantity increases, surface temperature of radiant components will increase further making the heat low. Thus, the difference of heat quantity of local radiant components must be accurately measured than heat variation rates of overall radiant quantity. Real radiant components, not a module with average heat flux and the heat quantity of each radiant component must be accurately predicted and analyzed. These data can be used to make a cold plate with a good performance.

5. Conclusions

This paper presented a contribution to a liquid cooling type in telecommunication equipment. An experimental investigation of the cold plate used in an amplifier module was performed. The results showed that when the circulation rate of a cooling liquid is 3 *l/min,* the 8 mm case shows a better performance than 10 mm case. Also, the surface temperature of electronic components in 8 mm flow passage shows a stable status and low temperature profiles. In the cooling plate with the same diameter for two geometries, the increase rate of the temperature in the heated surfaces shows lower than that of heat flux. And radiant quantity of electronic devices increases linearly when the temperature increases.

Acknowledgement

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