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Performance Comparison of Liquid-Cooling with Air-Cooling Heat Exchangers Designed for Telecommunication Equipment

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Abstract

Electronic and telecommunication industries are constantly striving towards miniaturization of electronic devices. Miniaturization of chips creates extra space on PCBs that can be populated with additional components, which decreases the heat transfer surface area and generates very high heat flux. Even though an air-cooling technology for telecommunication equipment has been developed in accordance with rapid growth in electrical industry, it is confronted with the limitation of cooling capacity due to the rapid increase of heat density. In this study, liquid-cooling heat exchangers were designed and tested by varying geometry and operating conditions. In addition, air-cooling heat exchangers were tested to provide performance data for the comparison with the liquid-cooling heat exchangers. The liquid-cooling heat exchangers had twelve rectangular channels with different flow paths of 1, 2, and 12. Silicon rubber heaters were used to control the heat load to the heat exchangers. Heat input ranged from 293 to 800W, and inlet temperatures of working fluid varied from 15 to 27℃. The heat transfer coefficients were strongly affected by flow conditions. All liquid-cooling heat exchangers showed higher cooling performance than the air-cooling heat exchanger. The heat exchanger with 2-paths could provide more controllability on the maximum temperature than the others.

Key words: Telecommunication equipment, Heat flux, Liquid cooling, Air cooling, Heat exchanger

Nomenclature

A_w : Wetted area in the rectangular channel [m^2]
D_h : Hydraulic diameter of channel [m]
$\frac{4A}{P_w}$
H : Heat transfer coefficient [W/m^2 ℃]
Q : Heat flux [W]
T : Temperature [℃]

Subscript

i : Inlet fluid
o : Outlet fluid
fr : Front side surface average
op : Opposite side surface average

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1. Introduction

Current telecommunication equipments installed in telecommunication shelters consist of rack mounted units with chips on PCB(printed circuit board). The information technology industry has been improved by increasing power density and heat dissipation within the footprint of telecommunications hardware. Since the electronic component is so weak for high temperatures, the maximum allowable temperature of the electronic component is limited. Normally, air-cooling and liquid-cooling technologies are applied to telecommunication equipments.

Mohamed [1] analyzed the air-cooling characteristics of an uniform square module and developed a heat transfer correlation. Sparrow [2] investigated the characteristics of a liquid-cooling heat exchanger for a uniform array of square modules. Jang [3] performed an experimental study on the heat transfer and pressure drop for an electronic cooling heat ex-

changer with mini-channels. Li et al.[4] reported that the flow and the heat transfer performance in a micro-channel were greatly affected by thermo-physical properties of the liquid. They also found that the bulk temperature along the flow direction could be represented in a quasi-linear form at high flow rate conditions. Bhowmik et al.[5] investigated the effect of heat fluxes, flow rates and geometrical parameters in a liquid cooled rectangular channel with discrete heat source. They developed experimental correlations for the Nusselt number based on experimental data.

Most of the previous researches on electronic equipment cooling focused air-cooling heat exchanger and a discrete heat source in micro scale. There were limited works on the liquid-cooling for the unit rack installed in the telecommunication shelter. In this study, the performance of the air-cooling unit rack for telecommunication equipments installed in the field was examined. In addition, the performance of liquid-cooling heat exchangers was measured by varying heat density and operating conditions.

2. Experimental apparatus

2.1 Experimental setup

Fig. 1 shows the schematic diagram of the experimental rig of a unit rack for telecommunication equipment. Inlet temperature of the test section was controlled by a constant temperature bath. The flow rate was set by adjusting a magnetic gear pump. Air-cooling heat exchanger (Fig. 2(a)) having an unit size of 420×330×140 mm^3 used flat plate type fins with the inner side pitch of 4 mm. Liquid-cooling heat exchangers (Fig. 2(b)), having volume of 215×300×16 mm^3 were made of plate aluminum. The rectangular channel size was 5.0×3.0 mm. Three liquid-cooling heat exchangers having twelve rectangular channels with different flow paths of 1, 2, and 12 were tested.

A silicon rubber heater instead of PCB was attached to the liquid-cooling heat exchangers to control heat input. The power to the silicon rubber heater was adjusted by a power supplier, and it was measured by a power meter with an uncertainty of ±0.01% of full scale (12 kW). T-type thermocouples were attached on the silicon rubber heaters and liquid-cooling heat exchangers to examine cooling performance and to check thermal distribution according to heat density as illustrated in Fig. 2. Liquid temperatures and pressures at the inlet and outlet of the heat exchanger were measured using RTD sensors, and pressure transducer, respectively.

The test rig was installed in a psychrometric calorimeter to maintain the environmental conditions of the telecommunication equipment. The psychrometric calorimeter consisted of a refrigerator, an electric heater, and a humidifier can control ambient temperature and relative humidity. Tests were executed by

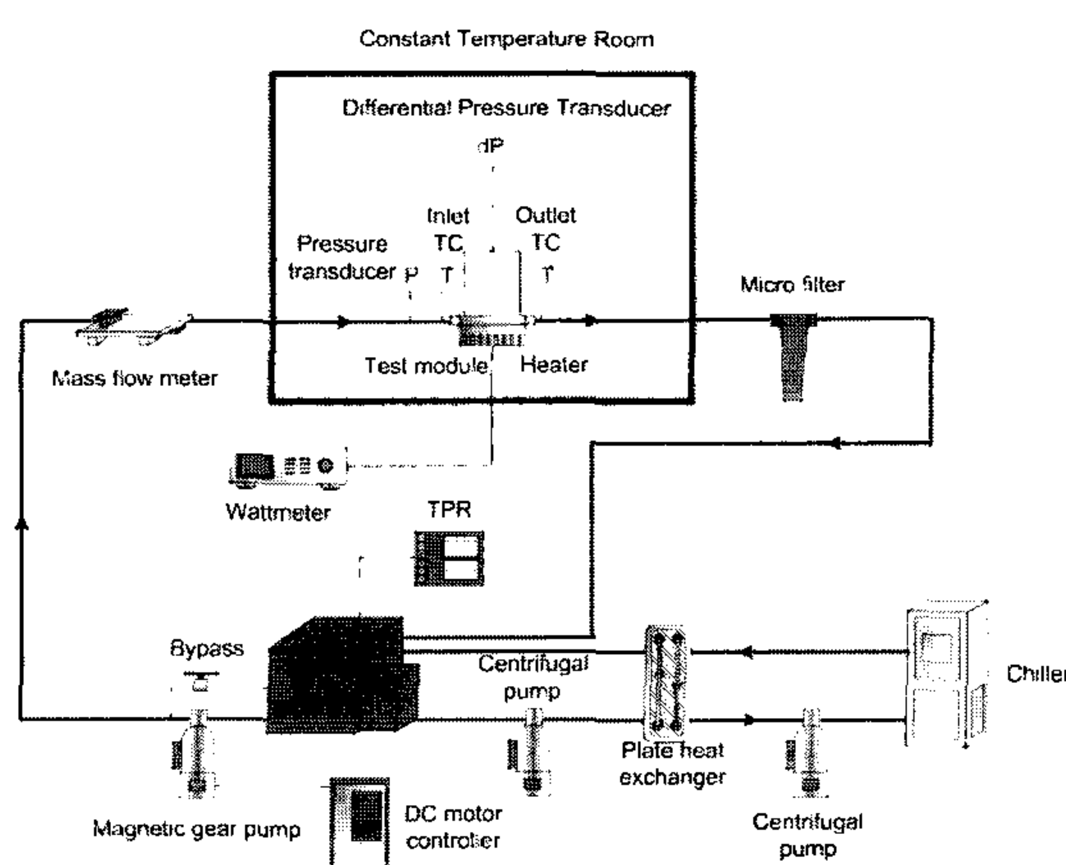
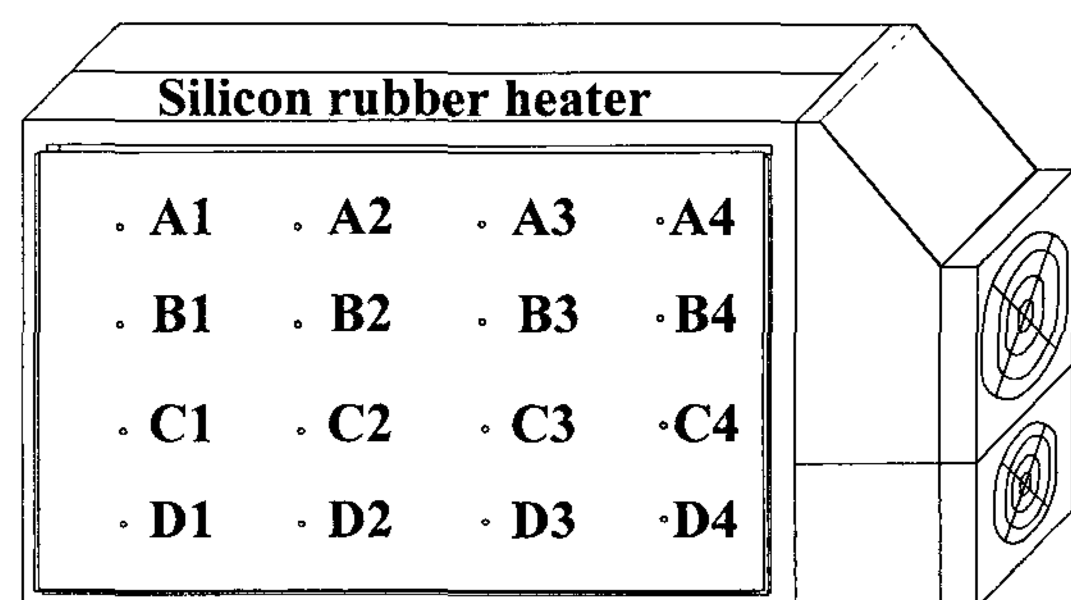
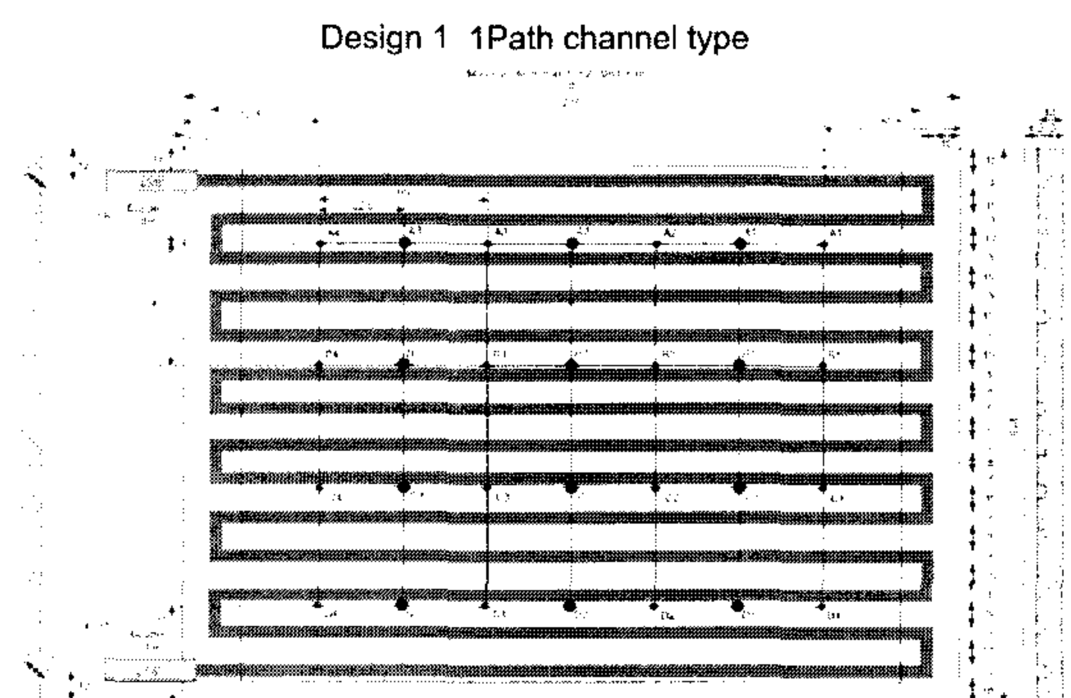


Fig. 1. Schematic diagram of the test rig.



(a) Air cooling heat exchanger



(b) Liquid cooling heat exchanger

Fig. 2. Location of TC on the tested heat exchangers.

Table 1. Test conditions.

Operating parameter	Range	Reference data
Inlet temperature of water, ℃	15 - 27	25
Mass flow rate of water, kg/h	11 - 244	50
Heat input, W	293 - 800	293 W
Ambient temperature	25℃, 50%RH	25℃, 50%RH

varying heat input to the silicon rubber heater, flow rate, and inlet temperature. Table 1 shows the test conditions in this study. The data were recorded every two second and averaged for a period of 20 minutes after steady state conditions were reached. Reference data were obtained from the cooling performance of the telecommunication equipment in the field.

2.2 Data reduction

The overall heat transfer coefficient, h, was defined by the temperature difference between the wall and fluid as given in Eq. (1) according to Lee et al. [6]. Thermal properties (specific heat, conductivity, viscosity, etc.) of the fluid were calculated using the average temperature between inlet and outlet of the test section. The equation considered the heat transmissivity and heat removing rate of the heat exchanger.

$$h = \frac{Q}{A_w\left(\frac{T_{fr}+T_{op}}{2} - \frac{T_i+T_o}{2}\right)} \quad (1)$$

2.3 Uncertainty analysis

Uncertainty evaluation for the heat transfer coefficient was performed with the data for 1 path heat exchanger. Total experimental uncertainty of the measurement was classified by bias error and precision error. Uncertainty of the heat transfer coefficient(h) was estimated from the following equations.

$$\frac{U_h}{h} = \sqrt{\left(\frac{B_h}{h}\right)^2 + \left(\frac{P_h}{h}\right)^2} \quad (2)$$

Bias limit :

$$\frac{B_h}{h} = \sqrt{\left(\frac{B_Q}{Q}\right)^2 + \left(\frac{B_{Aw}}{A_w}\right)^2 + \left(\frac{B_{\Delta T}}{\Delta T}\right)^2} \quad (3)$$

Table 2. Uncertainty analysis.

Error type	Analyzed value	Error (%)
Bias errors	h , heat transfer coefficient	2.072
	Q , heat input	0.01
	A_w , wetted area	2.00
	ΔT , temperature difference	0.54
Precision error	h , heat transfer coefficient	2.015
	h , heat input	0.187
	T , temperature	0.379
Uncertainty	h , heat transfer coefficient	2.890

Precision limit :

$$\frac{P_h}{h} = \sqrt{\left(\frac{P_Q}{Q}\right)^2 + \left(\frac{P_{T_{fr}}}{\Delta T}\right)^2 + \left(\frac{P_{T_{op}}}{\Delta T}\right)^2 + \left(\frac{P_{T_i}}{\Delta T}\right)^2 + \left(\frac{P_{T_o}}{\Delta T}\right)^2} \quad (4)$$

where, $P_{Aw} = 0$ because wetted area (A_w) is constant. Precision limit is defined by twice of standard deviation and calculated by the following equation.

$$P_x = 2S_x = 2\sqrt{\frac{1}{N-1}\sum_{k=1}^{N}(X_k - \overline{X})^2} \quad (5)$$

The results of uncertainty analysis are summarized in Table 2.

3. Results and discussion

Thermal characteristics of telecommunication equipment operated in the field were measured to provide the reference data for the experiments. The maximum air inlet and outlet temperatures of the rack were 25.4℃ and 30.4℃, respectively. Surface temperature of the PCB ranged from 35 to 60℃. The PCB surface temperatures at several locations were much higher than those of the air. This was due to the heat trap and improper air distribution of the telecommunication equipment. The maximum heat input that was consistent with the equipment capacity in the field was approximately 293 W. This value was used as a reference condition for the tests.

The maximum surface temperature difference in the liquid-cooling heat exchanger with one-path was lower than that in the air-cooling heat exchanger as shown in Fig. 3. The surface temperature difference decreased with the increase of liquid flow rate in the liquid-cooling heat exchanger, while it remained nearly constant in the air-cooling heat exchanger even

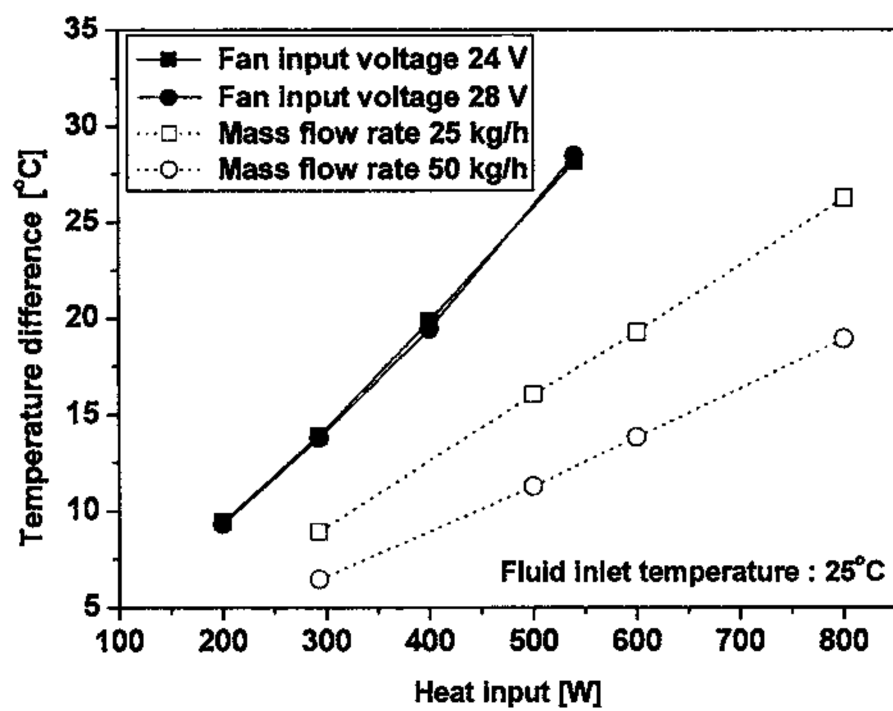


Fig. 3. Maximum surface temperature difference according to heat input.

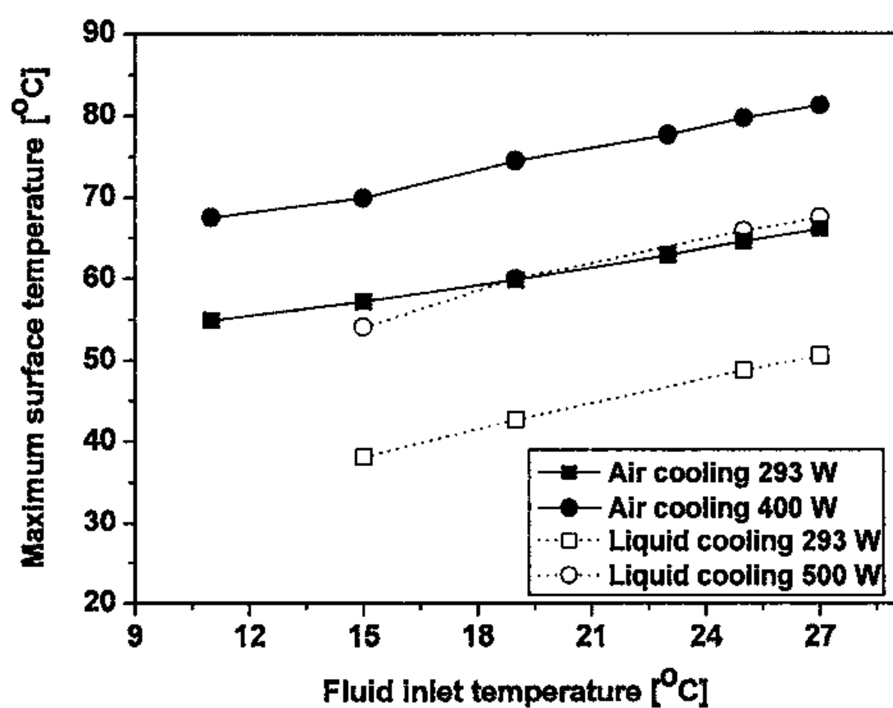


Fig. 4. Maximum surface temperature according to fluid inlet temperature.

though the air flow rate increased. This constant temperature difference in the air-cooling system was due to flow stagnation in the hot spot.

To get a thermal reliability for the telecommunication equipments, the maximum surface temperature of the PCB in the rack should be properly controlled. Fig. 4 shows the maximum surface temperature according to fluid inlet temperature for various heat inputs. The maximum surface temperatures for both heat exchangers linearly increased with the increase of the fluid inlet temperature. In addition, those values increased according to heat input to the silicon rubber heater with the increase of the cooling load. The maximum surface temperature in the liquid-cooling heat exchanger could be controlled at a lower value compared with that in the air-cooling heat exchanger even though the heat input for the former was higher than that for the latter. The maximum surface temperature of the liquid-cooling heat exchanger was lower than that of the air-cooling heat exchanger by approximately 15.5℃ at the heat input of 293 W for all fluid inlet temperatures. In addition, the size and the weight of the liquid-cooling heat exchanger were reduced by 83%, and 64%, respectively. This means that the liquid-cooling heat exchanger is more effective than the air-cooling heat exchanger to achieve thermal reliability of the telecommunication equipment.

Fig. 5 indicates the various surface mean temperature of the tested heat exchanger. The graph shows the almost same trend as the maximum surface temperature but the surface mean temperature of liquid cooling heat exchanger at the heat flux of 400 W was higher than the temperature of air cooling heat exchanger at the heat flux of 293 W. It means that the liquid cooling heat exchanger shows better ability of heat dissipation than the air cooling heat exchanger.

Fig. 6 represents the variation of the overall heat transfer coefficients, estimated by Eq. (1), with water flow rate for various flow paths in the liquid-cooling heat exchangers. The heat transfer coefficient increased with the increase of water flow rate because of the rise of the convective heat transfer rate. The flow in the one-path heat exchanger was in turbulent

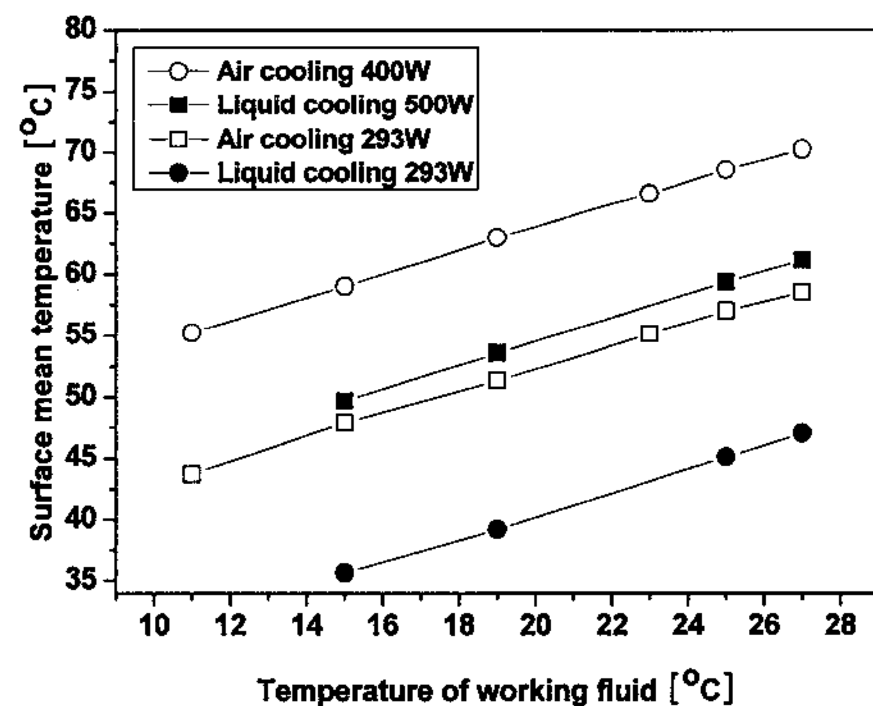


Fig. 5. Surface mean temperature according to fluid inlet temperature.

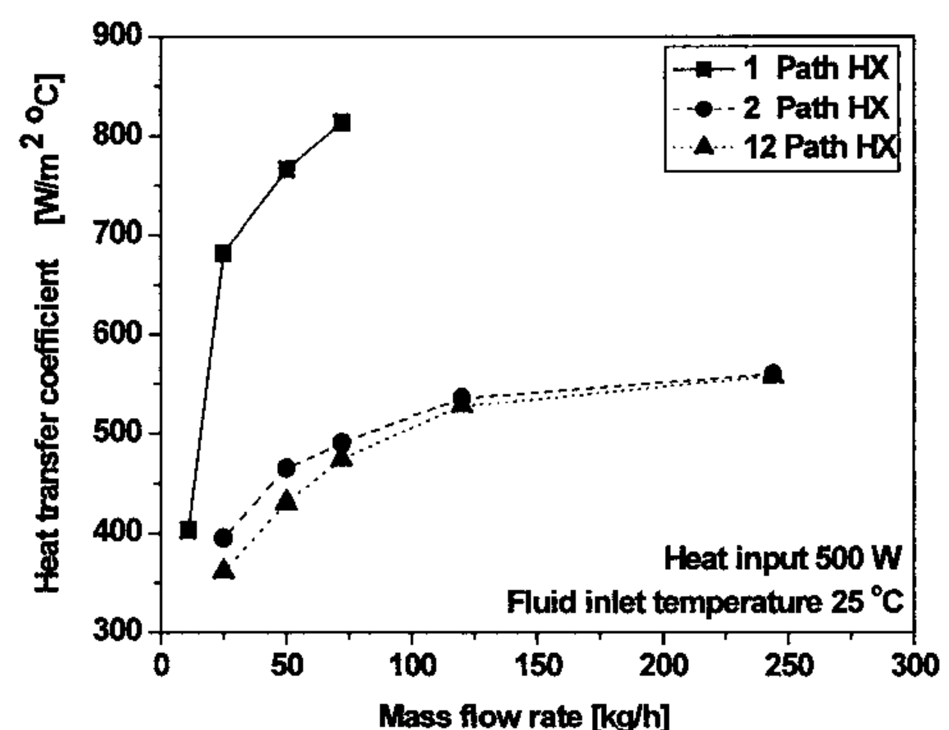


Fig. 6. Heat transfer coefficient according to mass flow rate for various flow paths.

flow, but it in the other geometries was in laminar flow. Therefore, the one-path heat exchanger showed the best performance among the tested heat exchangers. The heat transfer coefficient showed the same trends at all testing conditions.

Fig. 7 shows the variation of the maximum surface temperature as a function of mass flow rate for various flow paths in liquid-cooling heat exchangers. The maximum surface temperature of the two-path heat exchanger was lower than that of the other heat exchangers at all mass flow rates. For low flow rates, the surface temperature of the one-path heat exchanger was higher than that of the two-path heat exchanger even though the heat transfer coefficient of the former was higher than that of the latter. The outlet fluid temperature of the former was higher than that of the latter because of a longer flow path, resulting in the increase of the surface temperature at the outlet. The outlet fluid temperature decreased with the increase of flow rate, yielding the decrease of the surface temperature at the outlet. As a result, the maximum surface temperature for the both heat exchangers represented similar values at high flow rates. The maximum surface temperature of the twelve-path heat exchanger was higher than that of the others at all flow rates. This may be due to the mal-distribution of flow rate to multi paths.

The maximum surface temperature in the telecommunication equipment is the most important factor in the aspect of thermal reliability. The two-path liquid-cooling heat exchanger showed the best performance in the respect of heat transfer and pressure drop characteristics. Therefore, the two-path liquid cooling heat exchanger was recommended for cooling of the telecommunication equipment.

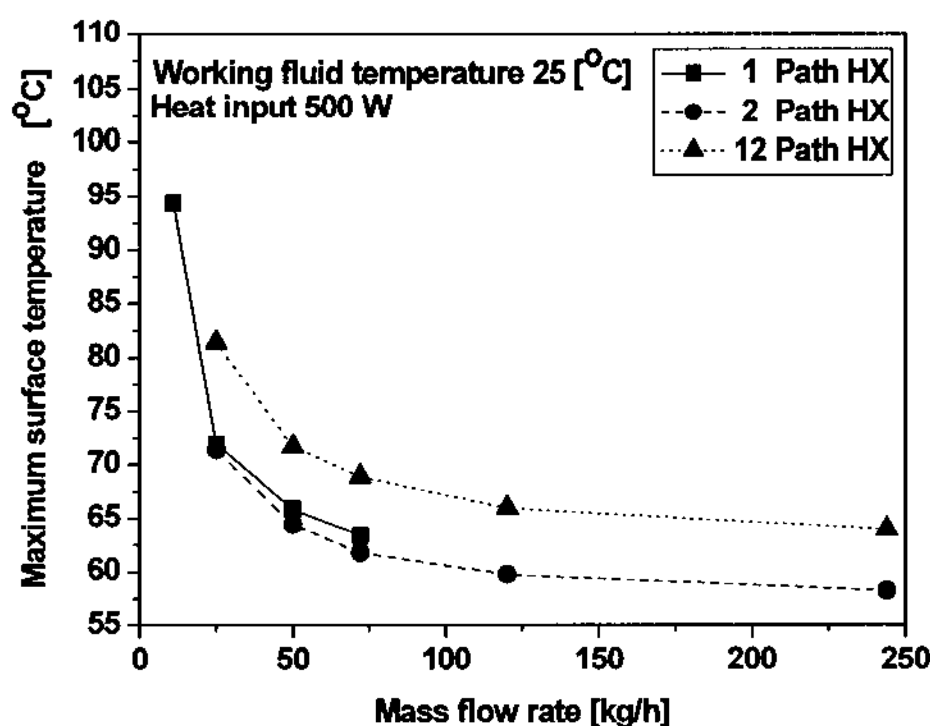


Fig. 7. Maximum surface temperature according to mass flow for various flow paths.

4. Conclusions

An experimental study on the performance of an air-cooling heat exchanger and liquid-cooling heat exchangers for telecommunication equipment was carried out. The maximum air inlet and outlet temperatures of the telecommunication equipment in the field were 25.4℃ and 30.4℃, respectively. The surface temperature of the PCB ranged from 35℃ to 60℃. The maximum surface temperature in the liquid-cooling heat exchanger could be controlled lower than that in the air-cooling heat exchanger even though the heat input for the former was higher than that for the latter. The maximum surface temperature in the liquid-cooling heat exchanger was lower than that in the air-cooling heat exchanger by approximately 15.5℃ at the heat input of 293 W at all fluid inlet temperatures. In addition, the size and the weight of the liquid-cooling heat exchanger were reduced by 83% and 64%, respectively. The two-path liquid-cooling heat exchanger showed the best performance among the tested heat exchangers in this study.

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