

A Loop Thermosyphon Type Cooling System for High Heat Flux

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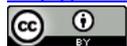
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Abstract

With rapid development of the semiconductor technology, more efficient cooling systems for electronic devices are needed. In this situation, in the present study, a loop thermosyphon type cooling system, which is composed mainly of a heating block, an evaporator and an air-cooled condenser, is investigated experimentally in order to evaluate the cooling performance. At first, it is examined that the optimum volume filling rate of this cooling system is approximately 40%. Next, four kinds of working fluids, R1234ze(E), R1234ze(Z), R134a and ethanol, are tested using a blasted heat transfer surface of the evaporator. In cases of R1234ze(E), R1234ze(Z), R134a and ethanol, the effective heat flux, at which the heating block surface temperature reaches 70°C, is 116 W/cm², 106 W/cm², 104 W/cm² and 60 W/cm², respectively. This result indicates that R1234ze(E) is the most suitable for the present cooling system. The minimum boiling thermal resistance of R1234ze(E) is 0.05 (cm²·K)/W around the effective heat flux of 100 W/cm². Finally, four kinds of heat transfer surfaces of the evaporator, smooth, blasted, copper-plated and finned surfaces, are tested using R1234ze(E) as working fluid. The boiling thermal resistance of the blasted surface is the smallest among tested heat transfer surfaces up to 116 W/cm² in effective heat flux. However, it increases drastically due to the appearance of dry-patch if the effective heat flux exceeds 116 W/cm². On the other hand, in cases of copper-plated and finned surfaces, the dry-patch does not appear up to 150 W/cm² in effective heat flux, and the boiling thermal resistances of those surfaces keep 0.1 (cm²·K)/W.

Keywords

Cooling, Boiling, Loop Thermosyphon, Global Warming Potential, Thermal Resistance

1. Introduction

Recent development of semiconductor technology brings the remarkable performance improvement and miniaturization of electronic devices. This leads to the increase of the heat flux dissipated from electronic devices. Therefore, the development of highly efficient cooling system for electronic devices is required in order to operate them normally.

Many researchers have been studied on various cooling systems for electronic devices such as air cooling systems, liquid cooling systems, heat pipe systems, etc. Mudawar [1] reviewed various cooling schemes such as pool boiling, detachable heat sinks, channel flow boiling, micro-channel and mini-channel heat sinks, jet-impingement, and sprays coping with high heat flux. Then, he reported that phase change played a key role in operating electric devices with high heat dissipation safely. He also pointed out it was important to predict critical heat flux (CHF) and to enhance it. There are many studies on the cooling systems applying the boiling phenomenon, in which the performance of systems are mainly evaluated with the thermal resistance and surface temperature of electronic devices to prevent the malfunction of electronic devices. Webb and Yamauchi [2] proposed air-cooled thermosyphon heat rejection device for CPU. Then, they examined the performance of the prototype made of aluminum using R134a (Global Warming Potential (GWP) = 1300) as the working fluid and reported that their system could support CPU heat rejection as large as 220 W. They also suggested that use of water (GWP = 0) as an environmentally friendly working fluid would meet possible future requirement. Gima *et al.* [3] investigated a closed loop thermosyphon type cooling system for high-power CPU experimentally. They tested four kinds of heat transfer surfaces of the evaporator, smooth flat surface, blasted flat surface, finned surface and blasted-finned surface, using FC-72 (GWP > 5000) as the working fluid, and they reported that blasted-finned surface showed the highest performance with the evaporator to ambient thermal resistance as 1.83 K/W ($45 \text{ cm}^2\text{-K/W}$) at heat input 30 W. Kawaguchi *et al.* [4] proposed a compact boiling refrigerant type cooling unit, which was miniaturized remarkably as compared with conventional air cooling fin unit, and conducted the performance test using R134a as the working fluid. They reported that the performance of the proposed cooling unit, 0.2 K/W ($1.25 \text{ cm}^2\text{-K/W}$) at heat load 200 W, was as high as that of conventional air cooling fin unit. They also proposed a calculation method to predict the performance of the proposed cooling unit. Matsushima and Usui [5] tested cooling characteristics of a thermosyphon type heat pipe using pure water and water/surfactant mixture; surfactant tested was sodium myristate. They reported that adding the surfactant 30 ppm to water decreased the total thermal resistance by 10% to 40%. In this case the smallest total thermal resistance was about 0.025 K/W ($4 \text{ cm}^2\text{-K/W}$) at heat input of 256 W. Chan *et al.* [6] developed a two-phase CPU cooler with a unique curvilinear fin condenser using water as the working fluid. They reported that the best performance of cooler thermal resistance was 0.206 K/W ($1.44 \text{ cm}^2\text{-K/W}$) at heat load 203 W and an air flow rate $0.98 \text{ m}^3/\text{min}$.

In order to improve performances of cooling systems applying the boiling phenomenon, enhancing the boiling heat transfer in evaporator is one of the most important issues. Parker and El-Genk [7] conducted experiments on the saturation and sub-cooled boiling of FC-72 on porous graphite and smooth copper surfaces (heating area: 1.0 cm^2). They reported that CHF increased linearly with liquid subcooling and CHF on porous graphite surface was higher than that of smooth copper surface; at liquid subcooling of 30 K, CHF on porous graphite was 57.1 W/cm^2 , while that of smooth copper was 29.5 W/cm^2 . They also compared their data of CHF with other researchers' data of copper surface, silicon surface, micro-finned silicon surface, etc. Mori and Okuyama [8] conducted experiments on the enhancement of the CHF in saturated pool boiling of water by attachment of a honeycomb-structured porous plate on a heated surface of 30 mm in diameter. They reported that the CHF reached 250 W/cm^2 as the height of honeycomb porous plate was decreased to 1.2 mm; this value of CHF was approximately 2.5 times higher than that of plane surface. They also proposed the CHF prediction model based on consideration of the mechanism of CHF enhancement by honeycomb-structured porous plate. Saiz Jabardo [9] reviewed previous studies on the surface structure effects on nucleate boiling and conducted experiments on the nucleate pool boiling of R134a and R123 (GWP = 79) on horizontal copper and brass tubes in order to confirm the surface roughness effects on the heat transfer characteristics. His experimental results showed that the heat transfer coefficient was influenced by not material but the reduced pressure, and that the optimum roughness changed with kinds of working fluid. El-Genk and Ali [10] investigated enhanced nucleate boiling on copper micro-porous surface layers using PF-5060 (GWP > 5000) as the working fluid. They tested five kinds of micro-porous surface layers (thickness: 95, 139, 171, 197 and 220 μm). They found that the optimum thickness

that made CHF maximum existed, and reported that the CHF (27.8 W/cm^2) of $171 \mu\text{m}$ thick layer was 17 or more times higher than those reported on plane surfaces. Li and Peterson [11] investigated the effects of geometric parameters of micro-porous coatings on the CHF. They tested 10 kinds of sintered pure copper woven mesh screens as micro-porous coatings on the heating block, and examined effects of the coating thickness, volumetric porosity and pore size on the CHF. They also pointed out that the capillary evaporation inside porous coatings might be one of the most important factors to enhance the CHF. Kwark *et al.* [12] created nanocoating surfaces by the nanofluid (Al_2O_3 -water/ethanol) pool boiling experiments, and measured their wetting and wicking characteristics. Then, they tested the pool boiling heat transfer characteristics of their nanocoating surfaces in pure water and found that the CHF of nanocoating developed in ethanol nanofluid (193 W/cm^2) was larger than that of nanocoating developed in water-based nanofluid (190 W/cm^2) due to the quasi-static contact angles of the nanocoating and surface wettability. Hendricks *et al.* [13] studied pool boiling heat transfer in water using nanostructured surfaces made of Zn Oon aluminum (Al) and copper (Cu) substrates. They reported that CHF using ZnO nanostructures on Al (contact angle 20°) was 82.5 W/cm^2 and it was larger than that (23.2 W/cm^2) of bare Al surface (contact angle 104°), and presumed that nanostructured surfaces created high nucleation site densities and bubble frequency and led to enhancement of CHF. Hosseini *et al.* [14] investigated experimentally roughness effects on nucleate pool boiling heat transfer of R113 (GWP = 5820) using four horizontal circular copper surfaces, the average roughness (Ra) of which were 0.901, 0.735, 0.695 and $0.090 \mu\text{m}$, respectively. Their results showed that the significant improvement of heat transfer coefficient as the average surface roughness increased; the heat transfer coefficient of Ra = $0.901 \mu\text{m}$ was 38.5% higher than that of Ra = $0.090 \mu\text{m}$ at heat flux of 17 W/cm^2 . Furberg and Palm [15] tested the boiling heat transfer of R134a and FC-72 on a dendritic and micro-porous surface. They measured bubble size, bubble frequency density and heat transfer coefficient and calculated the latent and sensible heat flux contributions in the heat flux range of 2 to 15 W/cm^2 . Then, they reported that the dendritic and micro-porous surface produced smaller bubbles and higher bubble frequency and it led to improvement of the boiling heat transfer coefficient. Jun *et al.* [16] carried out experiments on the pool boiling of ethanol and water on nano-textured surfaces, which were copper platelets covered with copper-platelectrospun nanofibers. They reported that the heat flux and heat transfer coefficient on the nano-textured surfaces were about 3 - 8 times higher than those on the bare copper surface. They also considered the effects of the nano-textured surfaces on the boiling heat transfer characteristics. McHale and Garimella [17] studied nucleate boiling of smooth and rough surfaces with FC-72. They tested 7 surfaces with different roughness and reported that active nucleation site densities were higher using the rough surface, and the vapor in contact with the rough surface did not spread as freely as on the smooth surface. They [18] also analyzed the heat transfer characteristics and bubble ebullition in nucleate boiling on the rough surface to clarify the roughness effects. As mentioned above, many previous studies attempted to increase CHF by processing the boiling surface structure, in order to correspond to the increase in the heat flux dissipated from electronic devices. Most of tested working fluids, however, have high GWP, and their use may be restricted in future.

Recently, new substances with extremely low GWP such as R1234ze(E), R1234ze(Z), were synthesized as environmentally acceptable working fluids. Therefore, this study aims to confirm whether these newly synthesized substances are suitable to use as working fluids of electronic device cooling systems. In this study, a loop thermosyphon type cooling system, which consists mainly of an evaporator and an air-cooled condenser, is investigated experimentally. Four type heat transfer surfaces in the evaporator are tested in order to assess the performance of the loop thermosyphon type cooling system using four kinds of working fluids. The heat transfer surfaces tested are smooth, blasted, copper-plated and finned ones, and the working fluids tested are R1234ze(E), R1234ze(Z), R134a and ethanol.

2. Experimental Method

2.1. Experimental Apparatus

Figure 1(a) shows a schematic diagram of an experimental apparatus used in the present study, and **Figure 1(b)** shows its photograph. This apparatus composed mainly of an evaporator and an air-cooled condenser is a loop thermosyphon type cooling system driven by gravity. The operating principle is as follows: the liquid of the working fluid is heated and boiled in the evaporator. After that the vapor of the working fluid rises along a vertical tube and enters the condenser. At the condenser, the vapor changes to the liquid. Then, the liquid from the condenser returns to the evaporator by the gravity. A heating block made of copper is attached upward to the

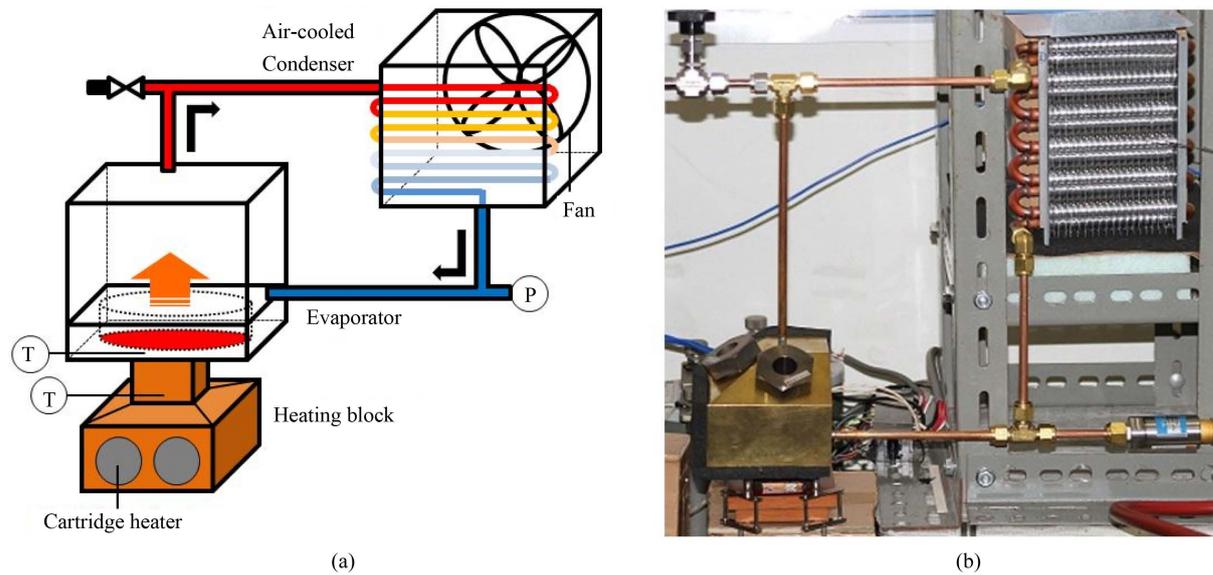


Figure 1. Experimental apparatus. (a) Schematic diagram; (b) Photograph.

bottom of the evaporator through thermal grease. Two cartridge heaters ($200\text{ W} \times 2$) inserted in the heating block generate the heat load to the evaporator. The heat load is controlled by a DC power supply. The charging process of the working fluid into the experimental apparatus is conducted by two steps mainly. At the first step, the experimental apparatus and the connecting hose between the experimental apparatus and the working fluid container are evacuated by an oil-sealed rotary vacuum pump. At the second step, the working fluid is introduced into the apparatus, weighing its mass.

Figure 2 shows four types of heat transfer surfaces used in the evaporator, where **Figures 2(a)-(d)** are the photographs of a smooth surface, a blasted surface, a copper-plated surface and a finned surface, respectively. All of the heat transfer surfaces are made of copper. The specifications of these heat transfer surfaces are shown in **Table 1**. The average surface roughness listed in **Table 1** is measured using the direct contact method or the laser speckle method. The values of the average surface roughness of smooth, blasted, copper-plated and finned surfaces are 7.75 , 3.52 , 40.8 and $0.68\ \mu\text{m}$, respectively. The blasted surface was made using sand particles in the diameter range of $53\ \mu\text{m}$ to $150\ \mu\text{m}$. The copper-plated surface was made by treating the smooth surface in three plating processes, the conditions of which are listed in **Table 2**. The finned surface has 45 square fins, and the dimension of each fin is as follows: the width is $1.0\ \text{mm}$, the length is $1.0\ \text{mm}$, the height is $4.0\ \text{mm}$, and the pitch between fins is $\sqrt{2}\ \text{mm}$.

Table 3 shows the experimental conditions. The experimental parameters are the type of heat transfer surface, the kind of working fluid, the volume filling rate, the average air velocity flowing to the condenser and the input heat flux. In the experiments, the liquid pressure at the inlet of the evaporator is measured with an absolute pressure transducer, the temperature distribution in the heating block is measured with three $\phi\ 1.0\ \text{mm}$ K-type sheathed thermocouples, the temperature inside the heat transfer surface is measured with a $\phi\ 1.0\ \text{mm}$ K-type sheathed thermocouple, the heat load of the cartridge heaters is measured with a DC-power meter, the average air velocity to the condenser is measured by an anemometer, and the ambient air temperature is measured with a $\phi\ 1.0\ \text{mm}$ K-type sheathed thermocouple.

2.2. Data Reduction

Figure 3 shows the model of performance evaluation indicators, that is, thermal resistances. In this figure, q_{eff} is the effective heat flux, T_{hb} , T_{eo} , T_{base} , T_f and T_a denote the heating block surface temperature, the bottom surface temperature of the evaporator, the heat transfer surface temperature, the saturation temperature of the working fluid and the ambient air temperature, respectively, and R_{CONT} , R_{EBW} , R_{BOIL} and R_{COND} are the contact thermal resistance of thermal grease between the heating block and the evaporator, the thermal resistance of the evaporator bottom wall, the boiling thermal resistance between the heat transfer surface and the working fluid, and the

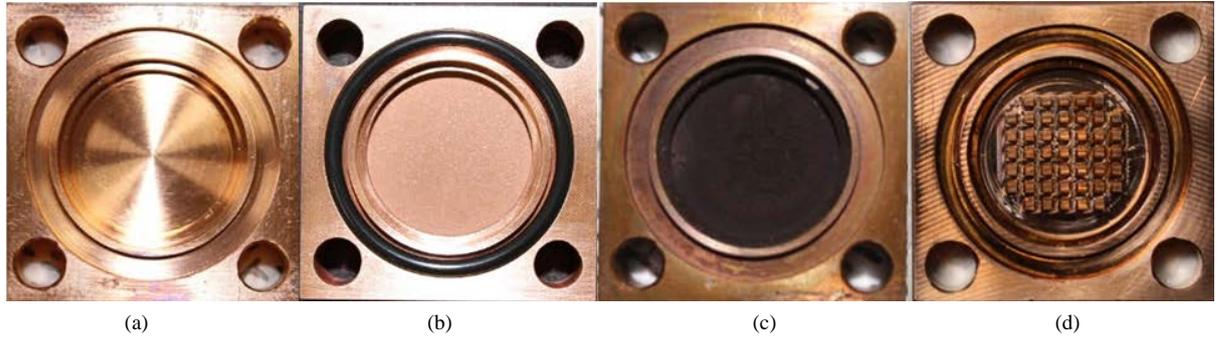


Figure 2. Heat transfer surfaces of evaporator. (a) Smooth; (b) Blasted; (c) Copper-plated; (d) Finned.

Table 1. Specifications of heat transfer surfaces in evaporator.

Surface	Surface Thickness (mm)	Inner Diameter of Heat Transfer Surface (mm)	Contact Area with Heating Block (mm ²)	Average Surface Roughness (μm)	Method*
Smooth	5.2	20	196 (14 × 14)	7.75	Laser
Blasted	5.2	20	196 (14 × 14)	3.52	Contact
Copper-plated	5.2	20	196 (14 × 14)	40.8	Laser
Finned	5.2	20	196 (14 × 14)	0.68	Contact

*method to measure average surface roughness

Table 2. Process of making the copper-plated surface.

Step	Current Density (mA/cm ²)	Current Value (mA)	Duration (min)
1	10	60.0	1
2	200	1200.0	5
3	5	30.0	2

Table 3. Experimental conditions.

Heat Transfer Surface	Smooth, Blasted, Copper-plated, Finned
Working Fluid	R1234ze(E), R1234ze(Z), R134a, Ethanol
Volume Filling Rate* (%)	20.2 - 72.4
Average Air Velocity (m/s)	1.4, 2.5
Input Heat Flux** (W/cm ²)	15 - 165

*Ratio of saturation liquid volume to internal volume of the system; **Calculated based on contact area between the evaporator and the heating block.

thermal resistance of condenser between the working fluid and the ambient air, respectively. In this study, q_{eff} , T_{hb} , T_{eo} and T_{base} are calculated from the temperature distribution in the heating block based on the assumption of the one dimensional heat conduction in the heating block and the evaporator bottom wall, and T_r is calculated from the measured pressure of the working fluid.

Thermal resistances to evaluate the heat transfer performance of the cooling system are defined as follows,

$$R_{CONT} = (T_{hb} - T_{eo})/q_{eff} \quad (1)$$

$$R_{EBW} = (T_{eo} - T_{base})/q_{eff} \quad (2)$$

$$R_{BOIL} = (T_{base} - T_r)/q_{eff} \quad (3)$$

$$R_{COND} = (T_r - T_a)/q_{eff} \quad (4)$$

The system performance of the cooling system is evaluated by the total thermal resistance R_{SYS} , expressed as,

$$R_{SYS} = R_{CONT} + R_{EBW} + R_{BOIL} + R_{COND} \quad (5)$$

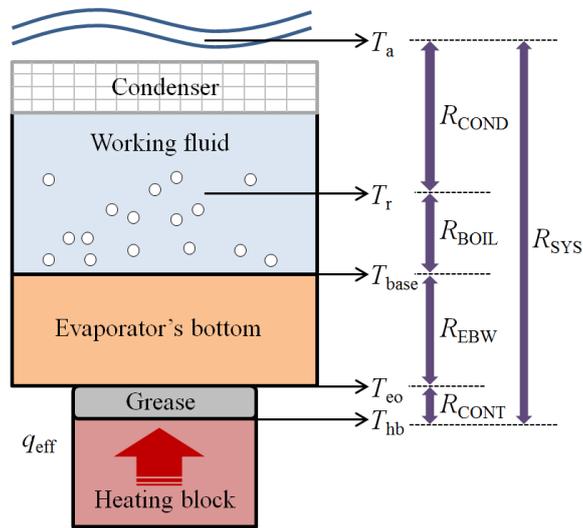


Figure 3. The model of evaluation indicator.

3. Results and Discussion

First, experiments were carried out in order to find the optimum volume filling rate of the working fluid. Then, effects of the average air velocity of condenser on thermal resistances were tested. The experimental results are compared in order to select the suitable working fluid and the highest performance heat transfer surface of the evaporator for the present cooling system.

3.1. Effects of Volume Filling Rate and Average Air Velocity of Condenser

Figures 4(a)-(b) show the effects of the volume filling rate on the boiling thermal resistance, R_{BOIL} , and the thermal resistance of condenser, R_{COND} , respectively, where the volume filling rate is defined as the ratio of the liquid volume to the total inner volume of the present cooling system at ambient air temperature 20°C. The experimental condition of these results is as follows: the working fluids is R1234ze(E), the heat transfer surface in the evaporator is smooth, and the average air velocity of condenser is 2.5 m/s. As shown in Figure 4(a), R_{BOIL} is affected little by the volume filling rate. On the other hand, as shown in Figure 4(b), R_{COND} is affected strongly by the volume filling rate. As the volume filling rate increases, the value of R_{COND} decreases once and reaches the minimum, and then it increases. The optimum volume filling rate in which R_{COND} is the minimum is about 40%. The optimum volume filling rate might be determined by the effective heat transfer area of the condenser and the circulating mass flow rate. On the other hand, the circulating mass flow rate, which is determined by the liquid head difference between the condenser and the evaporator, surges with the increase of the volume filling rate.

Figures 5(a)-(b) show the effects of the volume filling rate on R_{BOIL} and R_{COND} , respectively, in the case of the blasted surface. The experimental condition is the same as in Figure 4 except for the heat transfer surface. The result of the blasted surface is almost the same as that of the smooth surface shown in Figure 4. The optimum volume filling rate is about 40%. In the same manner, the optimum volume filling rate was determined for the other working fluids and the other heat transfer surfaces.

Figures 6(a)-(b) show the results of the breakdown of the system thermal resistance, R_{SYS} , in cases of the average air velocity of 1.4 m/s and 2.5 m/s, respectively. The other experimental conditions are as follows: the working fluid is R1234ze(E), the volume filling rate is about 40%, and the heat transfer surface is smooth. The boiling thermal resistance, R_{BOIL} , the thermal resistance of condenser, R_{COND} , the thermal resistance of the evaporator bottom wall, R_{EBW} , and the contact thermal resistance, R_{CONT} , are 15%, 40.7%, 24.7% and 19.6% at average air velocity of 1.4 m/s, respectively, while R_{BOIL} , R_{COND} , R_{EBW} and R_{CONT} are 18.6%, 33.3%, 26.8% and 21.3% at average air velocity of 2.5 m/s. From the comparison between Figures 6(a)-(b), it is found that only R_{COND} becomes smaller by increasing the average air velocity. This is due to the increase of air side heat transfer performance.

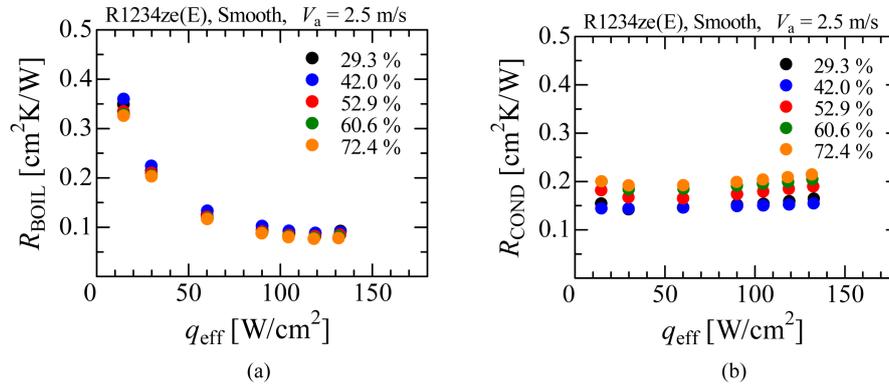


Figure 4. Effect of volume filling rate in case of smooth surface. (a) Boiling thermal resistance; (b) Thermal resistance of condenser.

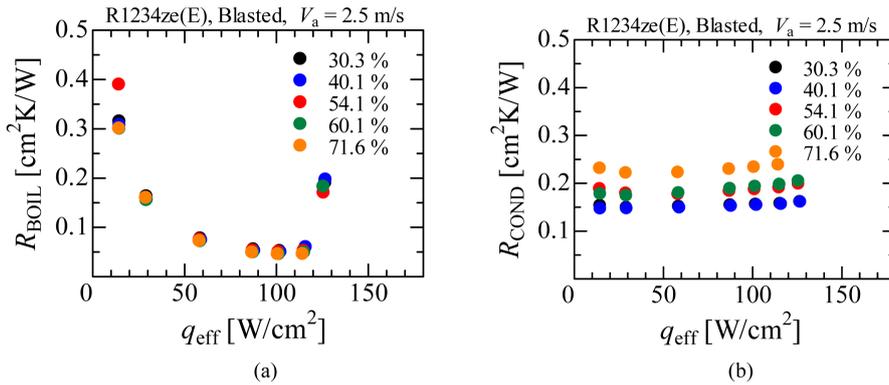


Figure 5. Effect of volume filling rate in case of blasted surface. (a) Boiling thermal resistance; (b) Thermal resistance of the condenser.

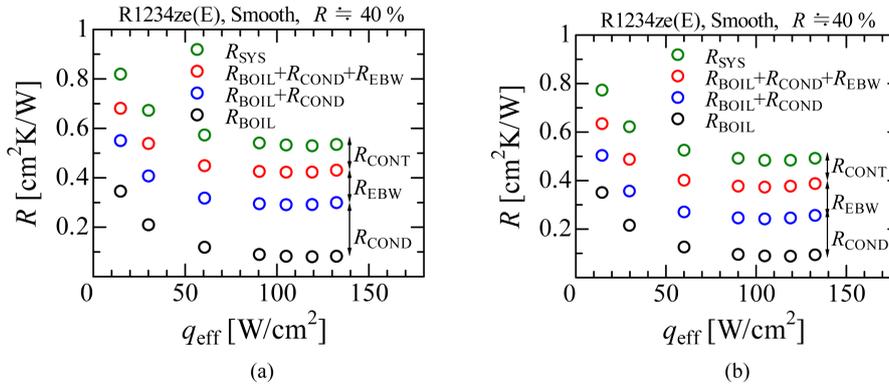


Figure 6. Effect of average air velocity of condenser on system thermal resistance. (a) Air velocity of 1.4 m/s; (b) Air velocity of 2.5 m/s.

3.2. Effect of Working Fluid on Cooling Performance

The experimental data of R1234ze(E), R1234ze(Z), R134a and ethanol are compared in order to select the suitable working fluid for the present cooling system. The data used in the comparison were obtained on the following experimental condition: the heat transfer surface is blasted one, the average air velocity of condenser is 2.5 m/s, and the volume filling rate is approximately 40%. The comparison results are shown in Figure 7.

Figure 7(a) shows the relation between R_{BOIL} and q_{eff} . In cases of R1234ze(E) and R1234ze(Z), R_{BOIL} decreases once with the increase of q_{eff} , and then it increases due to the appearance of dry-patch on the heat trans-

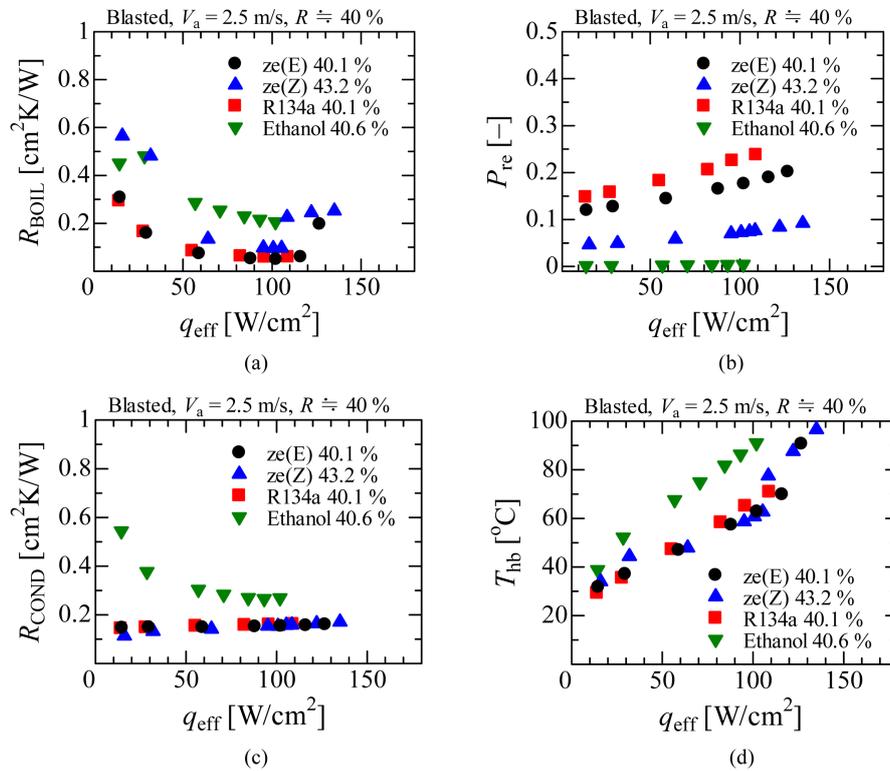


Figure 7. Effect of working fluid. (a) Boiling thermal resistance; (b) Reduced pressure; (c) Thermal resistance of condenser; (d) Heating block surface temperature.

fer surface. On the other hand, in cases of R134a and ethanol, R_{BOIL} decreases monotonously with the increase of q_{eff} . This is because the dry-patch does not appear in the present experimental ranges of R134a and ethanol. The minimum boiling thermal resistances of R1234ze(E), R134a, R1234ze(Z) and ethanol in the present experimental ranges are 0.05 ($\text{cm}^2\cdot\text{K}/\text{W}$), 0.06 ($\text{cm}^2\cdot\text{K}/\text{W}$), 0.1 ($\text{cm}^2\cdot\text{K}/\text{W}$) and 0.2 ($\text{cm}^2\cdot\text{K}/\text{W}$), respectively. R_{BOIL} is the reciprocal of the boiling heat transfer coefficient. In general, the boiling heat transfer coefficient is strongly related with the reduced pressure. It is observed that the higher the reduced pressure, the larger the heat transfer coefficient. As shown in **Figure 7(b)**, the reduced pressure at the same effective heat flux decreases in the following order: R134a, R1234ze(E), R1234ze(Z) and ethanol.

Figure 7(c) shows the relation between R_{COND} and q_{eff} . In the cases of R1234ze(E), R1234ze(Z) and R134a, R_{COND} increases a little with the increase of q_{eff} . On the other hand, in the case of ethanol, R_{COND} decreases with the increase of q_{eff} , and it is 1.5 or more times higher than in the cases of other working fluids.

Figure 7(d) shows the relation between the heating block surface temperature, T_{hb} , and q_{eff} . In all cases of working fluids, T_{hb} increases with the increase of q_{eff} . The effective heat flux, at which T_{hb} reaches 70°C , is as follows: 60 W/cm^2 , 104 W/cm^2 , 106 W/cm^2 and 116 W/cm^2 in cases of ethanol, R134a, R1234ze(Z) and R1234ze(E), respectively. This result indicates that R1234ze(E) is the most suitable for the present cooling system among test working fluids.

3.3. Effect of Heat Transfer Surface on Cooling Performance

The experimental data of smooth, blasted, copper-plated and finned surfaces are compared in order to find the suitable heat transfer surface of evaporator in the present cooling system. The data used in the comparison were obtained on the following experimental condition: the working fluid is R1234ze(E), the average air velocity of condenser is 2.5 m/s, and the volume filling rate is approximately 40%. The comparison result is shown in **Figure 8**.

Figure 8(a) shows the relation between R_{BOIL} and q_{eff} . In the cases of smooth, copper-plated and finned surfaces, R_{BOIL} gradually decreases with the increase of q_{eff} , and then it is kept approximately at 0.1 ($\text{cm}^2\cdot\text{K}/\text{W}$).

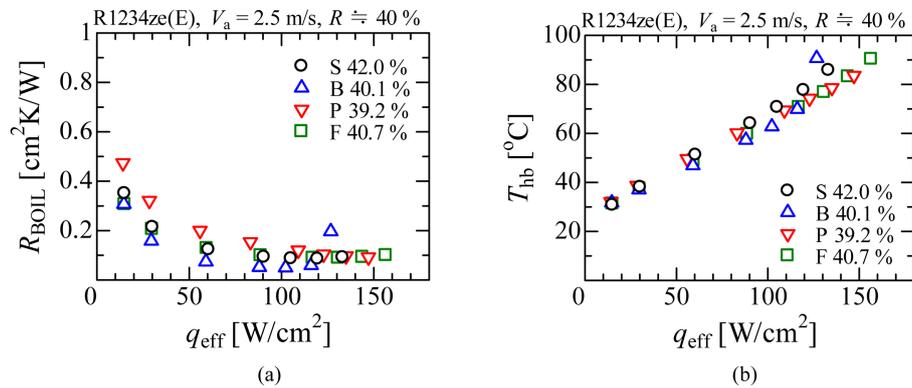


Figure 8. Effect of heat transfer surface. (a) Boiling thermal resistance; (b) Heating block surface temperature.

This characteristic corresponds to the transition from sub-cooled boiling to the saturated boiling. In the case of the blasted surface, the same characteristic is observed up to 116 W/cm² of the effective heat flux, and then R_{BOIL} increases drastically due to the appearance of dry-patch. In this case the minimum boiling thermal resistance is 0.05 (cm²·K)/W. The blasted surface shows the best performance up to 116 W/cm², but the dry-patch appears at the effective heat flux above 116 W/cm².

Figure 8(b) shows the relation between T_{hb} and q_{eff} . T_{hb} of all heat transfer surfaces increases with the increase of q_{eff} . The effective heat flux, at which T_{hb} reaches 70°C, is as follows: 102 W/cm², 115 W/cm², 110 W/cm² and 113 W/cm² in cases of smooth, blasted, copper-plated and finned surfaces, respectively. The effective heat flux, at which T_{hb} reaches 80°C, is as follows: 123 W/cm², 120 W/cm², 140 W/cm² and 135 W/cm² in cases of smooth, blasted, copper-plated and finned surfaces, respectively. As a result, it is found from **Figures 8(a)-(b)** that copper-plated and finned surfaces are suitable for the heat transfer surface of the evaporator.

4. Conclusions

The loop thermosyphon type cooling system for high heat flux was investigated experimentally in order to evaluate the cooling performance. The main findings are as follows.

- With the change in the volume filling rate, the thermal resistance of condenser is more greatly influenced than the boiling thermal resistance, and these indicate that the optimum volume filling rate is approximately 40%.
- The boiling thermal resistance, R_{BOIL} , the thermal resistance of condenser, R_{COND} , the thermal resistance of the evaporator bottom wall, R_{EBW} , and the contact thermal resistance, R_{CONT} , are 15%, 40.7%, 24.7% and 19.6% at average air velocity of 1.4 m/s, respectively, while R_{BOIL} , R_{COND} , R_{EBW} and R_{CONT} are 18.6%, 33.3%, 26.8% and 21.3% at average air velocity of 2.5 m/s. R_{COND} occupies the largest part in the system thermal resistance, and it becomes smaller as the average air velocity is larger.
- With the combination of R1234ze(E) and the blasted heat transfer surface of the evaporator, the boiling thermal resistance is the smallest up to 116 W/cm² of the effective heat flux, and the minimum boiling thermal resistance is 0.05 (cm²·K)/W around 100 W/cm² of the effective heat flux.
- With R1234ze(E), the boiling thermal resistance of the blasted surface drastically increases after the appearance of the dry-patch at 116 W/cm², while it of plated and finned surfaces is maintained as 0.1 (cm²·K)/W up to approximately 150 W/cm².

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