Optimal design of a micro evaporator to maximize heat transfer coefficient

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This paper presents an optimal design of a micro evaporator to maximize the heat transfer coefficient for an active micro cooler. The design of experiment methodology was applied for optimal design of the micro evaporator. The number of gaps, channel width and lateral gap size were selected as the design parameters and the power of heater was selected as noise factor. R-123 was used as the refrigerant and the flow rate was fixed as 0.72 g/s. The temperature at the surface of the heater was measured to valuate the heat transfer coefficient. A total of nine experiments were conducted using $L_9(3^4)$ orthogonal array. Intermediate optimal heat transfer coefficient was 0.395, 0.366 and 0.258 W/cm^{2}K and the heater power were at 40, 60 and 80 W, respectively, in the condition of 3 gaps, 0.5 mm channel width and 0.5 mm lateral gap size. Among the 3 design parameters, the channel width is the most sensitive parameter influencing the heat transfer coefficient. We conduct a second stage of experiment using two $L_4(2^3)$ array to increase the heat transfer coefficient by reselecting the channel width and the lateral gap size. Based on the experimental results, we could conclude that 3 gaps, 0.5 mm channel width and 1.25 mm lateral gap size are the final optimal design parameters for a micro evaporator resulting in the heat transfer coefficient of 0.465, 0.457 and 0.430 W/cm^{2}K for a heater power of 40, 60 and 80W, respectively.

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NOMENCLATURE

h = heat transfer coefficient
q = heat flux from the wall of evaporator to refrigerant
T_w =Temperature at the wall of evaporator
T_f= Temperature of the refrigerant
k_{al} = Conductivity of aluminum
t = Thickness of evaporator
$\dot{M}$ = Mass flow rate of refrigerant
C_p = Constant-pressure specific heat of refrigerant
T_{sat} = Saturation temperature of refrigerant
T_{in} = Temperature of refrigerant at inlet
A_L = The area of saturated liquid inside the evaporator

1. Introduction

Recently, the computer industry is advancing at a significant speed. Especially, the integration of CPU has become much denser than before. By Moore' law, CPU integration rate will increase by two times in every 18 months. As the CPU integration becomes denser, the heat from the CPU is also increased simultaneously. The heat from CPU is a critical factor influencing the quality and function of the computer operation. So, many industries and universities are trying their best to find ways to maintain the CPU in the proper temperature range.

Mostly, the conventional method to cool the CPU is using fan and fins. However, it is not sufficient to cool the CPU whose integration rate is becoming much denser. Particularly, inside a laptop computer, for example, that has limited space fans or fins are not a solution for the reason of size. Thus, there is currently a worldwide effort to find cooling methods to refrigerate CPU that emits more heat inside such limited space such as using heat transfer of phase change like boiling and condensing.

The method of spraying fluid to the heat source and the method of circulating the refrigerant for cooling the heat source is being widely researched. Both methods apply the heat transfer of phase change phenomena. Lian Zhang et al. have developed spray type cooling system that is integrated with semiconductor. It can be performed as 0.22 W/cm^{2}K in the condition of 2–3.5 ml/min flow rate [1]. Jelena Vukasinovic et al. have developed the small size (Φ152.4 mm) active radial countercurrent heat sink which can refrigerate 80 W of heat power at 70 °C [2]. S. I. Haider et al. have conducted research on closed-loop two phase thermosyphon that show function of 40 W/cm^2 heat transfer rate at 85°C by using wick type capillary phenomenon [3]. However, these kinds of passive cooling system cannot guarantee to refrigerate high density CPU, so development of an active cooling system that adopts refrigeration cycle is necessary in near future.

To develop an active micro cooler, in-depth research on the micro evaporator is needed. To analyze the micro evaporator, the research on two phase flow is performed widely. Kandlikar has suggested the analytical relation of the dimension and heat transfer inside the micro straight channel type evaporator, and have verified the relation by experiments [4, 5]. Wu et al. have designed a trapezoidal shape micro heat exchanger and have measured the phase change phenomena inside the channel [6]. Xu et al. have verified the relation between transient flow pattern and the heat transfer procedure by using triangular micro channel [7]. However, mostly, the previous researches are focused on the phenomena analysis on the one dimensional micro evaporator, which is not sufficient to increase the heat transfer performance in two-dimensionally shaped evaporator. In this research, the optimal design of two-dimensionally shaped evaporator was conducted to enhance the heat transfer performance. For optimal design, the design of experiment methodology was applied [8].

In Section 2, the conceptual design of micro active cooler and micro evaporator is presented. In Section 3, we set up the problem definition for optimal design. Section 4 presents the result of intermediate and final result of optimal design. Finally, concluding remarks are presented in Section 5.

2. Active micro cooler

2.1 Conceptual design

Active micro cooler system was designed to transfer heat from the heat source to environment by using vapor-compression refrigeration cycle. The cooler system is composed of evaporator, compressor, condenser and expansion valve. The size of cooler is designed as cylinder shape of Φ60 X 15. And R-123 is used as refrigerant. The composition and refrigeration cycle of active micro cooler is shown in Fig. 1 and stack shape conceptual design in Fig. 2.

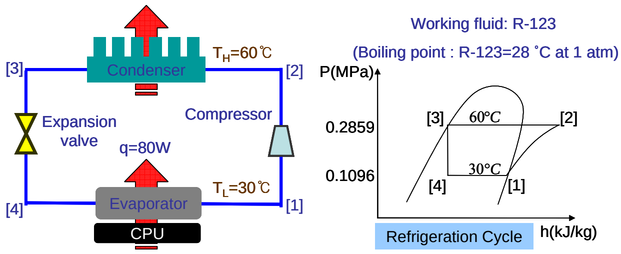

Fig. 1 Active refrigeration cycle

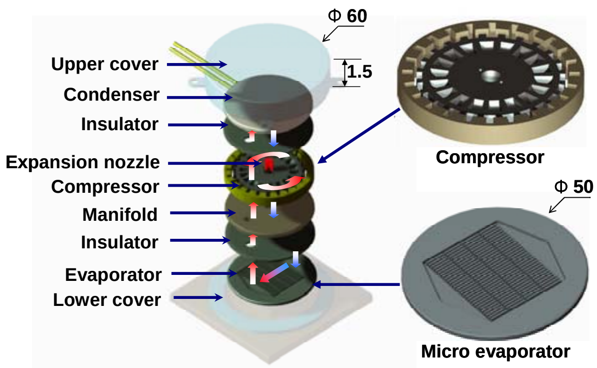

Fig. 2 Conceptual design of active micro cooler

2.2 Micro evaporator

To develop the active micro cooler, we have designed the micro evaporator. The micro evaporator is composed of several rectangular fins. The micro evaporator has cross links so the refrigerant can flow in two dimensional directions. The conceptual design of micro evaporator is shown in the right bottom side of Fig. 2.

3. Problem definition

3.1 Objective of optimal design

The performance of evaporator is valuated by the heat transfer coefficient and the pressure drop. It was found that the pressure drop is not a comparable factor during the experiment. Ideally, inside the evaporator, the pressure should be kept constant throughout the procedure. The variation of pressure drop among the evaporators is under 0.01 bar, so it is relatively much lower than the 1.1 bar which is pressure of evaporator outlet. So in this experiment, the pressure drop was not used to determine optimal design parameters. Only the heat transfer coefficient was used for optimal design.

3.2 Definition of objective value

We selected the heat transfer coefficient as the objective value for optimal design. The heat transfer coefficient can valuate the heat transfer rate from the wall of micro evaporator to refrigerant. The heat transfer coefficient is defined as follow:

$$h = \frac{q}{T_w - T_f} \tag{1}$$

where q is heat flux from the wall of evaporator and T_w and T_f are temperatures of the wall and the refrigerant, respectively. However, these T_w and T_f are not possible to measure directly in this experiment. So the heat transfer coefficient was calculated based on the relation from the measured values during the experiment.

First, the heat transfer equation from the heater to the micro evaporator is as follows:

$$T_w = T_h - \frac{qt}{k_{al}} \tag{2}$$

where T_h is temperature of heater, t is thickness of evaporator and k_{al} is the conductivity of the material (aluminum). By inserting the parameter values we can get $T_h - T_w = 0.073$ °C , so we can assume that T_h and T_w share the same value.

The area of saturated liquid inside the evaporator was calculated from the equation:

$$qA_L = \dot{M}C_p(T_{sat} - T_{in}) \tag{3}$$

where A_L is the area of saturated liquid inside the evaporator, $\dot{M}$ is the flow rate of refrigerant, C_p is constant-pressure specific heat of refrigerant, T_{sat} is the saturation temperature of refrigerant and T_{in} is the temperature of refrigerant at inlet. By inserting the parameter values, we can know the 95% of total area in the evaporator is full of saturated gas state. Then we can assume that the temperature of refrigerant is the same as the temperature of the saturated gas ($T_w = T_{sat}$).

Then we can get the relation between the objective value and the measured values as follows:

$$h = \frac{q}{T_h - T_{sat}}. \tag{4}$$

As the heat transfer coefficient is larger, the evaporator can transfer the heat from the heater to the refrigerant better. So, the objective of this research is to find optimal parameter set which can maximize the heat transfer coefficient.

3.3 Design parameters and constraints

We selected three parameters to be optimized: number of gaps, channel width and lateral gap size. Graphical explanation of each parameter are shown in Fig. 3. For the constraints, we fixed the height of fin as 1 mm. The whole size of the micro evaporator, material and type of refrigerant (R-123) were also fixed.

Table 1 Design parameters and level

	level 1	level 2	level 3
a (Number of gaps)	2	3	4
b (Channel width)	0.2 mm	0.35 mm	0.5 mm
c (Lateral gap size)	0.5 mm	0.75 mm	1.0 mm

Table 2 Intermediate experimental results based on orthogonal array $L_9(3^4)$ in the design of experiment

Experiment number	Design parameters			T_h (°C)			h (W/cm^{2}K)			S/N ratio (dB)
	a	b (mm)	c (mm)	40W	60W	80W	40W	60W	80W	
1-1	2	0.2	0.5	50.1	94.2	140.1	0.250	0.126	0.099	-41.03
1-2	2	0.35	0.75	41.5	59.7	108.1	0.407	0.262	0.139	-31.89
1-3	2	0.5	1.0	41.9	52.4	83.2	0.395	0.339	0.201	-25.88
1-4	3	0.2	0.75	65.7	104.0	135.2	0.147	0.109	0.104	-43.10
1-5	3	0.35	1.0	42.5	59.0	115.3	0.379	0.268	0.127	-33.19
1-6	3	0.5	0.5	41.9	50.6	71.0	0.395	0.366	0.258	-22.66
1-7	4	0.2	1.0	53.4	92.7	139.3	0.217	0.129	0.100	-41.07
1-8	4	0.35	0.5	40.5	54.7	88.5	0.439	0.310	0.183	-27.16
1-9	4	0.5	0.75	48.8	72.4	145.9	0.265	0.187	0.094	-39.49

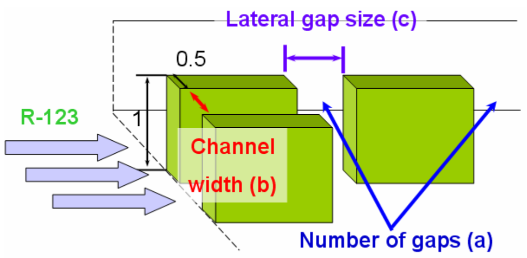

Fig. 3 Definition of the design parameters

The candidate of design parameters are shown in Table 1. To select the optimal parameter set, we conducted nine experiment by using $L_9(3^4)$ orthogonal array. As the noise factor, we changed the power of heater to 40, 60 and 80W. The orthogonal array is presented in the left side of Table 2.

4. Optimal design by experiment

4.1 Intermediate result of optimal design

4.1.1 Micro evaporator

Along to the $L_9(3^4)$ orthogonal array, nine micro evaporators were manufactured. The nine micro evaporators are shown in Fig. 4. The material used was Al 7075 and the whole size is 50 mm X 50 mm X 1.5 mm.

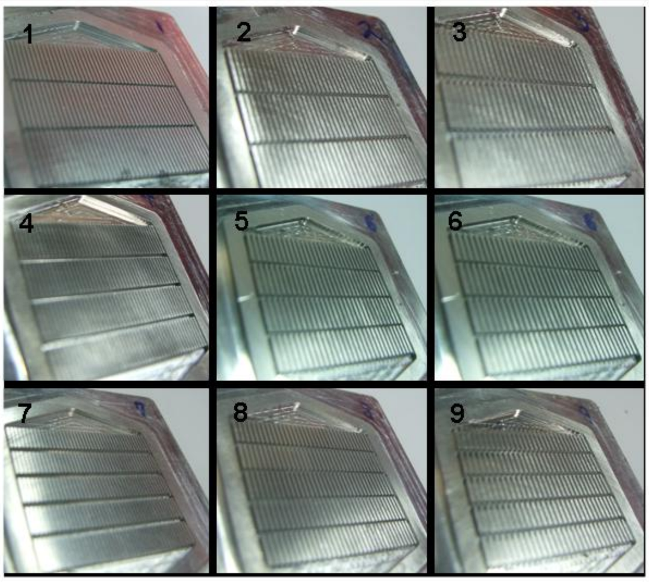

Fig. 4 Picture of micro channels

4.1.2 Experimental set up

The experimental set up shown in Fig. 5 is equipped for optimal design. The syringe pump supplies the saturated liquid state refrigerant as constant flow rate. The refrigerant enters the housing and passes the evaporator. Inside the evaporator, the refrigerant changes the phase to gas state. We measured the temperature of the heater for optimal design, and we also measured the temperatures at inlet and outlet of housing to check the phase change. Pressure drop was also measured for confirmation.

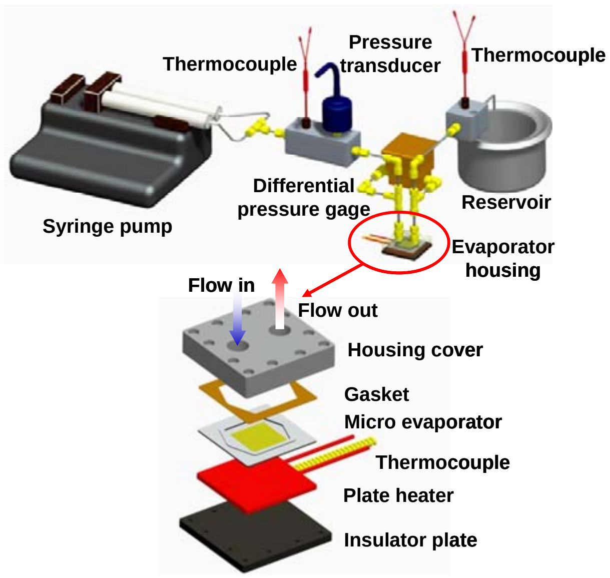

Fig. 5 Experimental set up

4.1.3 Result of the optimal design

Along to the $L_9(3^4)$ array, you can find the result of nine experimental result in the right side of Table 2. To calculate the heat transfer coefficient, we used the constant values as follows:

$$T_{sat}(R-123) = 27.85°C \quad (5)$$
$$Area = 720mm^2$$

We analyzed the sensitivity of parameters by using S/N ratio (Signal-to-Noise ratio). The definition of larger-the-better S/N ratio is as follows:

$$S/N = -10\log\frac{\left|\sum_{i=1}^{n} 1/y_i^{\,2}\right|}{n} \quad (6)$$

where y_i is the measured output and n is the numbers of experiment.

The experimental result is shown in Fig. 6. The intermediate optimal parameter set was 3, 0.5 mm and 0.5 mm, respectively. Among the three parameters, the channel width was determined to be the most sensitive parameter to the heat transfer coefficient.

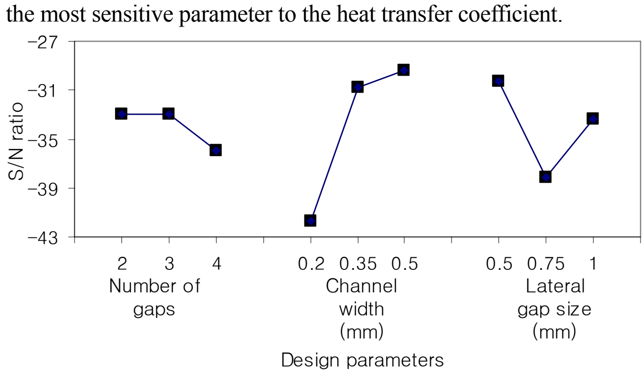

Fig. 6 Sensitivity analysis of each design parameters

Table 3 Final experimental results based on two tries of orthogonal array $L_4(2^3)$ in the design of experiment

Experiment number	Design parameters			T_h (°C)			h (W/cm^{2}K)			S/N ratio(dB)
	a	b (mm)	c (mm)	40W	60W	80W	40W	60W	80W	
2-1	3	0.5	1.0	41.0	48.3	62.1	0.423	0.408	0.324	-19.52
2-2	3	0.5	1.25	39.8	46.1	53.7	0.465	0.457	0.430	-15.98
2-3	3	0.65	1.0	41.0	47.9	56.3	0.423	0.416	0.391	-17.89
2-4	3	0.65	1.25	40.1	46.8	54.1	0.454	0.440	0.423	-16.50
2-5	3	0.5	0.25	44.2	52.8	62.7	0.340	0.334	0.319	-22.14
2-6	3	0.5	0.5	41.4	48.1	59.1	0.410	0.412	0.356	-18.85
2-7	3	0.65	0.25	40.4	47.5	58.3	0.443	0.424	0.365	-18.01
2-8	3	0.65	0.5	40.6	47.1	53.8	0.436	0.433	0.428	-16.78

4.2 Final result of the optimal design

Based on the result, we designed the second stage of experiment. We designed the experiment to modify the channel width and the lateral gap size to enhance the heat transfer coefficient. We used two set of $L_4(2^3)$ array because the tendency of the lateral gap size showed a bi-directional increase. The design of second experiment is shown in the left side of Table 3.

The experimental result is shown on the right side of Table 3. From the S/N ratio analysis, the final optimal set are 3, 0.5 mm and 1.25 mm, respectively. The final optimal S/N ratio and the intermediate optimal S/N ratio have a difference of 6.68 dB.

The deviation of the second stage of experiment is 6.16 dB, which is relatively lower than that of the deviation of first stage of experiment, 20.44 dB. It means that the heat transfer coefficient almost reach the maximum at the second stage of the experiment. So we can decide that the result of the second stage of experiment is the optimal parameter set that maximize the heat transfer coefficient.

5. Conclusion

In this paper, we designed optimal parameters to maximize the heat transfer coefficient. Selected parameters are the number of gaps, the channel width, the lateral gap size. The optimal parameter set is determined as 3, 0.5 mm and 1.25 mm respectively. The maximized heat transfer coefficients are 0.465, 0.457 and 0.430 W/cm^{2}K for the heater powers of 40, 60 and 80 W, respectively. From the sensitivity analysis, the channel width is the most sensitive parameter among the design parameters. We are planning to perform additional experiment to visualize the phase change phenomena inside the evaporator to figure out the reason of heat transfer enhancement.

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