Fundamental Experiments and Numerical Analyses on Heat Transfer Characteristics of a Vapor Chamber*  
(Effect of Heat Source Size)

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A vapor chamber is used as a novel heat spreader to cool high-performance MPUs (microprocessor units). The vapor chamber is placed between small heat sources and a large heat sink. This paper describes the effect of heat source size on the heat transfer characteristics of the vapor chamber. First, by the experiments, the effect of heat source size on the temperature distribution of the vapor chamber is investigated, and the validity of the mathematical model of the vapor chamber is confirmed. Secondly, by the numerical analyses, the effect of heat source size on the thermal resistances inside the vapor chamber is discussed. It is found that the heat source size greatly affects the thermal resistance of the evaporator section inside the vapor chamber. Although the thermal resistance is hardly affected by the heat generation rate and the heat flux of the heat source, it increases as the heat source becomes smaller.

Key Words: Heat Transfer, Heat Pipe, Thermosyphon, Vapor Chamber, Heat Spreader, Heat Source Size

1. Introduction

In recent years, a flat-plate type heat pipe called “Vapor Chamber” is widely used as a novel heat spreader to cool high-performance MPUs (microprocessor units) in personal computers, workstations and servers. The MPUs are high-heat-flux heat sources of small sizes, and the vapor chamber is placed between the small MPUs and a large heat sink to spread the heat from the former to the latter, resulting in the low thermal resistance between them. The principle of operation of the vapor chamber is basically the same as a conventional cylindrical heat pipe. The latent heat of evaporation and condensation of the working fluid is utilized and two-dimensional heat transfer is allowed inside the vapor chamber. Effectiveness of the vapor chamber has been already confirmed(1),(2), and the other advantages of the vapor chamber are that it is fabricated easily with low cost, the thickness is less than 5 mm, and it is lighter than the solid copper heat spreader of the same size.

The current vapor chamber works effectively without encountering any problems, but the improvement of the thermal performance of the vapor chamber is indispensable for cooling next generation MPUs. This is because the heat flux dissipated from the MPUs is increasing year-by-year resulting from the progress of data processing performance and the downsizing of the MPUs. In view of the above-mentioned fact, some experimental studies were already carried out on the heat transfer characteristics of the vapor chamber, and the useful data were presented for the improvement of the vapor chamber(3)–(6). Moreover, a mathematical model of the vapor chamber was developed to analyze the thermal-fluid transport phenomena occurring inside the vapor chamber and to predict the thermal performance of the vapor chamber in a short calculation time(7)–(9).

This paper describes the effect of heat source size on the heat transfer characteristics of the vapor chamber. From previous studies of conventional heat pipes, it is considered that the heat source size is one of the important parameters to dominate the heat transfer characteristics of the vapor chamber; nevertheless, very few stud-
ies have been published in this respect. Moreover, this kind of study is required currently because the downsizing of the MPUs is ongoing. First, by using three kinds of heat sources having different sizes, the fundamental experiment is carried out to investigate the effect of heat source size on the temperature distribution of the vapor chamber. Secondly, by using the mathematical model of the vapor chamber developed by the authors(7)–(9), numerical analysis is moreover carried out to discuss the effect of heat source size on the thermal resistances inside the vapor chamber. Similar to conventional cylindrical heat pipes, the thermal resistance of the vapor chamber is divided into two parts of evaporator and condenser sections inside the vapor chamber. Each of them is discussed in this paper.

2. Nomenclature

\[ A : \text{area \ cm}^2, \ \text{m}^2 \]
\[ c_p : \text{specific heat J/(kg \cdot K)} \]
\[ h_{fg} : \text{latent heat of evaporation and condensation J/kg} \]
\[ K : \text{permeability \ m}^2 \]
\[ k : \text{thermal conductivity W/(m \cdot K)} \]
\[ p : \text{pressure Pa} \]
\[ Q : \text{heat generation rate W} \]
\[ q : \text{heat flux W/cm}^2, \ \text{W/m}^2 \]
\[ R : \text{thermal resistance K/W} \]
\[ r : \text{radial coordinate mm, m} \]
\[ r' : \text{radius mm, m} \]
\[ T : \text{temperature} \ ^\circ\text{C} \]
\[ \langle T \rangle : \text{average temperature} \ ^\circ\text{C} \]
\[ V : \text{velocity vector} = (v, w) \text{ m/s} \]
\[ v : \text{radial velocity m/s} \]
\[ w : \text{axial velocity m/s} \]
\[ z : \text{axial coordinate mm, m} \]
\[ \varepsilon : \text{porosity} \]
\[ \mu : \text{viscosity Pa \cdot s} \]
\[ \rho : \text{density kg/m}^3 \]

Subscripts

1, 2, ..., 5 : positions indicated in Fig. 2
air : cooling air
b : bottom
c : condenser section
cal : calculated
e : evaporator section
eff : effective
exp : experimental
l : liquid-wick region
s : solid wall region
sat : saturated
sink : heat sink
sou : heat source
t : top
v : vapor region
ve : vapor chamber

3. Vapor Chamber

Figure 1 shows a photograph of the vapor chamber mounted on the base plate of the heat sink. This is the same vapor chamber as described in the authors’ previous paper(5). This paper addresses the extended study of this vapor chamber on the effect of heat source size. The vapor chamber is essentially a flat-plate type heat pipe made of copper. Water is enclosed as the working fluid. Sinter sheets (thickness: 0.5 mm) and sinter columns (diameter: 8.5 mm) made of small copper powders (powder size: 100 – 200 mesh, porosity: 40%) are used as the wick to circulate the working fluid. The sinter sheets are attached to the upper and lower internal surfaces, and then the sinter columns are set inside the vapor chamber to ensure the circulation of the working fluid.

The vapor chamber is placed between small heat sources and a large heat sink. Inside the vapor chamber, the working fluid receives the heat from the heat source and evaporates. The generated vapor spreads and flows to the upper internal surface, where it condenses, and the condensate returns to the lower internal surface through the wick to continue the circulation of the working fluid. Based on this principle of operation, the vapor chamber spreads the heat continuously from the heat source to the heat sink.

4. Experimental Apparatus and Procedure

Figure 2 shows a schematic illustration of the experimental apparatus. It consists of a wind tunnel, a heat sink with a vapor chamber, a heater used as a heat source, a blower, an orifice manometer setup and a damper. The wind tunnel is 890 mm long and its cross section is rectangular (height: 31.0 mm, width: 78.0 mm). The heat sink is placed in contact with the internal surface of the wind tunnel so as not to cause the airflow deviation along the gap between the heat sink and the wind tunnel. The vapor chamber is mounted on the base plate of the heat sink.
The bottom surface area of the heat sink (= 76.2 mm × 88.9 mm) is equal to the top and bottom surface areas of the vapor chamber, and they are much larger than the heating surface area of the heater (see Table 1). The heater is in the form of a square block and is attached to the center of the bottom of the vapor chamber, where the thermal grease (thermal conductivity: 0.64 W/(m·K)) is applied. The damper is used to control the air velocity. The wind tunnel, the vapor chamber and the heater are covered with the thermal insulators made of foamed plastic and glass wool to minimize any heat loss from them to the atmosphere.

In the experiment, three kinds of heaters, H1, H2 and H3, having different heating surface areas, Asou, as shown in Table 1, are used. The heat generation rate, Q, is changed as Q = 24, 48, 72 W, · · · , while the air temperature and the apparent air velocity inside the wind tunnel, which is 0.73 of the pore velocity in the heat sink, are maintained at 25°C and 1.5 m/s, respectively. The temperature distribution of the vapor chamber in a steady state condition is measured by K-type thermocouples. The temperature at the center of the bottom of the vapor chamber in contact with the heater, Tb, is measured by a thermocouple fixed in a small groove machined on the top of the heater. The temperatures at the top of the vapor chamber, Tt1, Tt2, · · · , Tt5, are measured by five thermocouples located at the positions indicated in Fig. 2. Five small holes are drilled into the base plate (thickness: 2.5 mm) of the heat sink and the thermocouples are inserted into them. The air velocity is measured by the orifice manometer setup.

Similar to conventional heat pipes, the vapor chamber also has a maximum heat transfer rate. If Q is inputted over this limitation, a part of the wick in the evaporator section inside the vapor chamber dries out, resulting in the degradation of the thermal performance of the vapor chamber. As mentioned above, the present experiment is carried out by increasing Q, but to avoid this problem, it is stopped before Q reaches the maximum heat transfer rate of the vapor chamber. The heat input from the heater is measured by a wattmeter, while the heat output from the heat sink inside the wind tunnel is calculated from the airflow rate and the temperature difference between the inlet and outlet of the heat sink. Because the measured heat input agrees within 5.5% with the calculated heat output, the former value is adopted as Q.

### Table 1 Specification of the heaters

<table>
<thead>
<tr>
<th>Type</th>
<th>Asou [cm²]</th>
</tr>
</thead>
<tbody>
<tr>
<td>H1</td>
<td>1.5 (1.22 cm²)</td>
</tr>
<tr>
<td>H2</td>
<td>3.0 (1.73 cm²)</td>
</tr>
<tr>
<td>H3</td>
<td>6.0 (2.45 cm²)</td>
</tr>
</tbody>
</table>

The mathematical modeling and numerical solution procedure

Because the details of the mathematical model of the vapor chamber and its numerical solution procedure were already contained in the authors’ previous papers[7]–[9], only their summaries are presented here.

Figure 3 shows the mathematical model of the vapor chamber, which consists of three regions of a vapor, a liquid-wick and a solid wall. For simplicity, the vapor chamber is modeled as an axisymmetric disk-shaped one and the numerical analysis is carried out in a cylindrical coordinate by using the following equivalent radii, rv, rsw.

\[
rv = \left(\frac{Avc}{\pi}\right)^{1/2} \quad (1)
\]

\[
rsou = \left(\frac{Asou}{\pi}\right)^{1/2} \quad (2)
\]

where Avc is the top surface area (= bottom surface area) of the actual vapor chamber, and Asou the heating surface area of the actual heat source (= contact area of the actual heat source at the bottom of the vapor chamber). The heat flux, q, is applied from the heat source and the top of the vapor chamber is entirely cooled by the heat sink.

The following assumptions:

a) A steady state condition is established inside the vapor chamber
b) The vapor flow is laminar
c) The wick is isotropic
d) The circulation of the working fluid is ensured
e) The vapor condenses and the liquid evaporates only at the interface between the vapor and liquid-wick regions
Fig. 3 Mathematical model and boundary conditions

Table 2 Governing equations

For the vapor region,
\[ \nabla \cdot \mathbf{V}_v = 0 \]  
(3)
\[ \rho_v \mathbf{V}_v \cdot \nabla \mathbf{V}_v = -\nabla p_v + \mu \nabla^2 \mathbf{V}_v \]  
(4)
\[ \rho_v c_p \mathbf{V}_v \cdot \nabla T_v = k_v \nabla^2 T_v \]  
(5)
For the liquid-wick region,
\[ \nabla \cdot \mathbf{V}_l = 0 \]  
(6)
\[ \rho_l \mathbf{V}_l \cdot \nabla \mathbf{V}_l = -\nabla p_l + \mu \nabla^2 \mathbf{V}_l - \varepsilon \mu \mathbf{V}_l / k \]  
(7)
\[ \rho_l c_p \mathbf{V}_l \cdot \nabla T_l = (k_{eff} / \varepsilon) \nabla^2 T_l \]  
(8)
For the solid wall region,
\[ \nabla^2 T_s = 0 \]  
(9)

At \([#1]\) : \( \mu_v (\partial \mathbf{V}_v / \partial z) = -\mu_w (\partial \mathbf{V}_w / \partial z) \), \( \rho_w \mathbf{V}_w = \rho_0 \mathbf{V}_w - k_v (\partial T_v / \partial z) \) \( \partial T_v / \partial r \) \( p_v \), \( T \) \( T_{sat} \), \( T_{air} \)
At \([#2]\) : \( \mu_v (\partial \mathbf{V}_v / \partial r) = -\mu_w (\partial \mathbf{V}_w / \partial r) \), \( \rho_w \mathbf{V}_w = \rho_0 \mathbf{V}_w - k_v (\partial T_v / \partial r) \) \( \partial T_v / \partial r \) \( p_v \), \( T \) \( T_{sat} \)

are made in conducting the analysis, and the governing equations for the vapor, the liquid-wick and the solid wall regions are given respectively as shown in Table 2, where \( \mathbf{V} \) is the velocity vector \( (v, w) \), \( p \) the pressure, \( T \) the temperature, \( \rho \) the density, \( \mu \) the viscosity, \( c_p \) the specific heat, \( k \) the thermal conductivity, \( \varepsilon \) the porosity, and \( K \) the permeability. Subscripts \( v \), \( l \) and \( s \) stand for the vapor, the liquid-wick and the solid wall regions, respectively. Equations (6) – (8) are derived by using the pore velocity, and \( k_{eff} \) is the effective thermal conductivity in the liquid-wick region.

The boundary conditions are also indicated in Fig. 3, where \( h_{fg} \) is the latent heat, \( T_{air} \) the temperature of cooling air. The temperature along the vapor-liquid interface is taken as a saturated temperature, \( T_{sat} \), corresponding to the pressure inside the vapor chamber. The authors have the experimental result\(^{(5)}\) on the heat transfer characteristics of the heat sink shown in Fig. 1, so the value of the thermal resistance of the heat sink, \( R_{sink} \), is obtained from them (\( R_{sink} = 0.249 \text{ K/W} \)).

The control volume method and the SIMPLE algorithm\(^{(10)}\) are employed to solve the governing equations with the boundary conditions. Because \( T_{sat} \) cannot be determined from the above equations, the numerical results are obtained with \( T_{sat} \) as a parameter, and then the one result is selected based on the following steady-state energy balances within 0.1%: the evaporative heat transfer rate inside the vapor chamber agrees with the rate of condensation, and the heat input to the vapor chamber agrees with the heat output from the vapor chamber.

The specification of the mathematical model is determined as shown in Table 3 based on the actual vapor chamber shown in Fig. 1.

### Table 3 Specification of the vapor chamber

<table>
<thead>
<tr>
<th>Material (chamber)</th>
<th>Material (wick sheet and wick column)</th>
<th>Working fluid</th>
<th>Height of vapor chamber</th>
<th>Radius of vapor chamber, ( r^*_{wc} )</th>
<th>Height of vapor region</th>
<th>Thickness of wick sheet</th>
<th>Radius of wick column</th>
</tr>
</thead>
<tbody>
<tr>
<td>Copper</td>
<td>Sinter made of copper powders (porosity: 40%)</td>
<td>Water</td>
<td>4.6 mm</td>
<td>46.4 mm</td>
<td>1.1 mm</td>
<td>0.5 mm</td>
<td>4.3 mm</td>
</tr>
</tbody>
</table>

6. Results and Discussion

6.1 Temperature distribution inside the vapor chamber

Figure 4 shows the experimental results of the temperature at the center of the bottom of the vapor chamber, \( T_b \), and the temperatures at the top of the vapor chamber, \( T_{1,1}, T_{1,2}, \ldots, T_{1,5} \). \( T_{1,5} \) is found to be slightly lower than \( T_{1,1}, T_{1,2}, T_{1,3} \) and \( T_{1,4} \) because it is located near the air inlet of the heat sink. However, the small differences in \( T_{1,1}, T_{1,2}, \ldots, T_{1,5} \) implies that the heat applied from the heater to the vapor chamber is spread almost uniformly to the heat sink. In the following discussion, the average of \( T_{1,1}, T_{1,2}, \ldots, T_{1,5} \) is used and expressed as \( <T> \). Figure 5 shows the
Fig. 4 Temperatures at the top and bottom of the vapor chamber

Fig. 5 Relation between temperatures of the vapor chamber and heat generation rate

experimental results of the relation between $T_b$, $\langle T_i \rangle$, and $Q$. It is confirmed that $T_b$ is greatly affected by the heat source size, $A_{soa}$, while the difference is not observed in $\langle T_i \rangle$.

Concerning $T_b$ and $\langle T_i \rangle$, the experimental results, $T_{b,exp}$, $\langle T_{i,exp} \rangle$, are compared in Fig. 6 with the numerical results, $T_{b,cal}$, $\langle T_{i,cal} \rangle$. The agreement between $\langle T_{i,exp} \rangle$ and $\langle T_{i,cal} \rangle$ is very close because the value of the thermal resistance of the heat sink, $R_{sink}$, is obtained from the authors’ previous experimental results as mentioned above. The agreement between $T_{b,exp}$ and $T_{b,cal}$, on the other hand, is known to depend on the estimation of the effective thermal conductivity, $k_{eff}$, in the liquid-wick region, and in the present numerical analysis, the excellent agreement is obtained as shown in the figure when $k_{eff} = 4.0 \text{W} / (\text{m} \cdot \text{K})$ is given. This value of $k_{eff}$ is considered to be valid because the published equations by Chi and Yagi-Kunii, which were both presented for estimating $k_{eff}$ of sintered metal powder wicks, give $k_{eff} = 3.36 - 3.61 \text{W} / (\text{m} \cdot \text{K})$ and $k_{eff} = 7.93 - 8.40 \text{W} / (\text{m} \cdot \text{K})$ respectively, and the present value of $k_{eff} = 4.0 \text{W} / (\text{m} \cdot \text{K})$ lies between them. This value of $k_{eff}$ is used in the following further calculations.

Figure 7 shows the numerical results of the temperature distribution inside the vapor chamber. The dashed lines represent the interfaces between the vapor and liquid-wick regions, and the liquid-wick and solid wall regions. In addition, the numerical result of the vapor velocity distribution inside the vapor chamber is shown in Fig. 8. Inside the vapor chamber, the liquid in the liquid-wick region receives the heat from the heat source and evaporates. The generated vapor spreads through the vapor region and then condenses at the upper interface between the vapor and liquid-wick regions. The condensate returns from the upper to the lower wick to continue the circulation of the working fluid. Because the latent heat of evaporation and condensation of the working fluid is utilized to transport the heat, it is found that the temperatures in the vapor re-
and therefore at the top of the vapor chamber are almost uniform. The numerical results also indicate that the temperature gradient near the heat source becomes larger as \( A_{\text{so}} \) decreases, because the decrease in \( A_{\text{so}} \) causes to increase the heat flux, \( q \), from the heat source; while the temperature at the top of the vapor chamber is hardly affected by \( A_{\text{so}} \), which is similar to the experimental results shown in Figs. 4 and 5. Because the effective thermal conductivity in the liquid-wick region is considerably low compared to the thermal conductivity in the solid wall region, the temperature gradient in the former region is much larger than that in the latter region.

For the cases of Fig. 7(a) and (b), the heat transfer rates through the bottom surface (\( r = 0 \)–4.3 mm, \( z = 2.0 \) mm) of the centered wick column are calculated at 10.4 W and 5.1 W, respectively. This heat transfer rate is found to increase as \( A_{\text{so}} \) decreases. However, because both the values are much smaller than the heat input, \( Q = 144 \) W, it is confirmed that the heat flow through the centered wick column is very small. Most of the heat applied from the heat source is transferred through the vapor region utilizing the latent heat of evaporation and condensation of the working fluid.

### 6.2 Thermal resistance inside the vapor chamber

The thermal resistance of the vapor chamber, \( R_{\text{vc}} \), is defined by the following equation:

\[
R_{\text{vc}} = (T_b - <T_\text{t}>) / Q \tag{10}
\]

and, by using the saturated temperature, \( T_{\text{satur}} \), corresponding to the pressure inside the vapor chamber, \( R_{\text{vc}} \), is divided into two parts, namely, the thermal resistances of the evaporator section, \( R_e \), and the condenser section, \( R_c \), inside the vapor chamber. \( R_e \) and \( R_c \) are expressed respectively as

\[
R_e = (T_b - T_{\text{satur}}) / Q \tag{11}
\]

\[
R_c = (T_{\text{satur}} - <T_\text{t}>) / Q \tag{12}
\]

The heat generation rate, \( Q \), in Eqs. (10)–(12) is expressed as \( Q = q \times A_{\text{so}} \), and the effects of \( q \) and \( A_{\text{so}} \) on \( R_{\text{vc}} \), \( R_e \), and \( R_c \) are examined numerically in Fig. 9. Irrespective
Effect of heat source size on thermal resistances for $Q = 72$ and 144 W

Effect of centered wick column on thermal resistances of $q$ and $A_{sour}$, it is found that $R_e$ is very small and $R_r$ is nearly equal to $R_{vc}$, implying that $R_r$ is dominant inside the vapor chamber. The numerical results also indicate that $R_e$ and $R_{vc}$ are hardly affected by $q$, but they increase with the decrease in $A_{sour}$. Figure 10 shows the numerical result of the relation between $R_e$, $R_{vc}$ and $A_{sour}$. It is also revealed that even when $Q$ is constant, $R_e$ and $R_{vc}$ increase as $A_{sour}$ decreases.

6.3 Effect of centered wick column

The numerical results are moreover obtained when the centered wick column inside the vapor chamber is removed, and the effect of centered wick column on $R_{vc}$, $R_r$ and $R_e$ is shown in Fig. 11. The purpose of the centered wick column is to ensure the circulation of the working fluid and to resist the external forces squashing the vapor chamber. Irrespective of $A_{sour}$, it is found that $R_e$ and $R_{vc}$ with the centered wick column are about 13% higher than those without the centered wick column. This is because $k_{eff}$ is comparatively low and the centered wick column stops the evaporation of the working fluid at the vapor-liquid interface at $r = 0$ - 4.3 mm, $z = 2.0$ mm. Because the small amount of heat is transferred through the centered wick column as mentioned above, a slight decrease in $T_{sat}$ is observed when the centered wick column is placed. $R_e$ is very sensitive to this change in $T_{sat}$ due to the very small value of ($T_{sat} - <T_i>$), and therefore $R_e$ with the centered wick column is about 10% lower than that without the centered wick column. Although the vapor chamber used in this study has the centered wick column above the heat source, it is better to place it at the position except just above the heat source inside the vapor chamber to decrease the thermal resistance and to increase the thermal performance of the vapor chamber.

7. Conclusions

Experimental and numerical analyses were carried out to investigate the effect of heat source size on the heat transfer characteristics of the vapor chamber, which is used as a novel heat spreader for cooling high-heat-flux MPUs. It was found that the thermal resistance of the condenser section inside the vapor chamber was very small, and that of evaporator section was nearly equal to the total thermal resistance of the vapor chamber, implying that the thermal resistance of the evaporator section was dominant inside the vapor chamber. It was also revealed that the thermal resistance of the vapor chamber was hardly affected by the heat flux applied from the heat source, while it increased as the heat source became smaller. The wick column is placed inside the vapor chamber to ensure the circulation of the working fluid and to resist the external forces squashing the vapor chamber. However, the numerical results also showed that irrespective of the heat source size, the thermal resistance of the vapor chamber was increased by 13% by the centered wick column placed just above the heat source inside the vapor chamber. Therefore, the wick column should be placed at the position except just above the heat source inside the vapor chamber to decrease the thermal resistance and to increase the thermal performance of the vapor chamber.

References


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