THE EFFECT OF THICKNESS AND PERMEABILITY OF WICK STRUCTURE ON L-SHAPE HEAT PIPE PERFORMANCE USING DIFFERENT WORKING FLUIDS

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ABSTRACT

This paper presents a two-dimensional FEM simulation of working fluid behavior inside L-shape heat pipe to predict wall temperature and liquid pressure under natural and forced convection modes by using ANSYS-FLOTTRAN. This work aimed to numerically study the effect of thickness and permeability of wick structure using different working fluids on L-shape heat pipe performance, where water and methanol were used as working fluids. The heat inputs for natural and forced convection modes comprised 20 W and 35 W respectively whereas the heat pipe wick thicknesses were taken as 0.5 mm and 0.75 mm. The results showed that at forced convection the use of water as working fluid with 0.5 mm sintered copper wick recorded the smallest temperature difference (9.8 °C) while methanol with 0.75 mm screen mesh wick recorded the largest temperature difference (28.27 °C).

Keywords: *L-shape heat pipe; ANSYS-FLOTTRAN; Wick structure; Working fluid; Liquid pressure drop.*

1. INTRODUCTION

Effective cooling of electronic components is important for the successful functioning and high reliability of electronic devices. The rapid developments in microprocessors necessitate enhanced processing power to ensure faster operations. According to (Shao and Zhang 2011), "Local hot spot heat fluxes within the CPU are exceeding 100W/cm² causing the CPU temperature rise, malfunction and failure of CPU without a very effective heat transfer device." Consequently, the requirement for cooling techniques to dissipate the associated heat is of utmost importance. Moreover, the present standard metallic heat sinks are obsolete in many ways and are not sufficient to address the increased cooling needs that are sought by many of today's electronic devices. Hence, it is highly desirable to explore high-performance cooling devices, especially for CPU cooling. Heat pipe has been identified and proved as one of the viable and promising options to achieved this purpose, due to its simple structure, flexibility and in particular, high efficiency.

Substantial numerical and experimental works have been reported on the application of heat pipes for electronic cooling. Reviews by (Faghri 1994), (Groll et al. 1998) and (Vasiliev 2005) provided detailed information on this topic. In addition, the main theoretical and numerical studies on heat pipes with capillary grooved structures were investigated by (Khrustalev and Faghri 1995a; b), (Faghri and Khrustalev 1997), (Khrustalev and Faghri 1999), (Zhang and Faghri 2001), (Jacolot et al. 2008), (Do et al. 2008), (Do and Jang 2010).

Two dimensional models that analyze the liquid and vapor flow and wall temperature along horizontal cylindrical heat pipe are developed by (Mistry et al. 2010), (Nouri-Borujerdi and Layeghi 2004), (Mahjoub and Mahtabroshan 2008). As well, some researchers investigated the flat heat pipes experimentally and numerically such as (Wang and Vafai 2000), (Maziuk et al. 2001), (Carbajal et al. 2007).

The differences in the shape of heat pipe affect its performance as the behaviors of the fluid and the wick structure inside the heat pipe play important role in the transmission of heat. However, the previous works

on numerical characterization by taking into account the behavior of the working fluid considered only conventional horizontal cylindrical heat pipe.

In the present study, a two-dimensional FEM simulation of working fluid behavior inside L-shape heat pipe was performed to predict liquid pressure and the wall temperature. The complexity of the simulation of the present heat pipe lies in the fact that the evaporator is at x direction while the condenser is at y direction. The simulations were done by using ANSYS-FLOTTRAN software and were validated by experimental results. The main objective of this work is to numerically study the effect of thickness and permeability of wick structure using different working fluids on L-shape heat pipe performance as water and methanol were used as working fluids.

2. MATHEMATICAL FORMULATION

The finned flat heat pipe under investigation is shown in Fig. 1. This finned heat pipe serves to cool the CPU of a modern notebook such as ASPIRE4720Z with Intel Pentium dual-core processor T2390, 1.86GHz and memory up to 4 GB DDR2. In this system, the heat pipe is used to transfer heat from CPU to the fins in a remote location, usually in the sides or corners of the notebook PC. The finned flat heat pipe consists of copper base plate of size 30 mm × 30 mm and condenser section of 72 mm long, with 50 aluminum rectangular fins of 20 mm × 10 mm size. The system is supported by radial fan with dimensions of 73 mm × 73 mm × 10 mm (width × length × thickness). The specifications of the heat pipe are shown in Fig. 1 and summarized in Table 1.

In this study, the adapted mathematical model was based on the following assumptions: vapor and liquid flows are assumed to be in steady state, laminar and incompressible; the vapor is treated as ideal gas; there is no heat generation due to phase change and chemical reaction in the system; the liquid and vapor phases are coupled at the liquid-vapor interface, and the vapor injection and suction at the liquid vapor interface are uniform; the physical properties are constant.

(a) Cross-section of flat heat pipe and equivalent (original) diameter (OD)

(b) Thermocouples locations

Fig.1. Finned flat heat pipe (a) Heat pipe cross-section (b) Thermocouples locations

Table 1 Specifications of finned L-shape heat pipe

Characteristic	Dimensions/Material
Evaporator length L_e	30 mm
Condenser Length L_c	72 mm
Adiabatic length L_{ad}	110 mm
Flat heat pipe	7.6 mm × 3.4 mm (width × thickness)
Original diameter (OD)	6 mm
Heat pipe Container thickness	0.5 mm
Wick thickness	0.5 mm
Working fluid	Water
Wick structure	Sintered copper powder
Material of pipe	Copper

The resulting governing equations are listed as follows:

Vapor region

$$\text{Continuity: } \frac{\partial u_v}{\partial x} + \frac{\partial v_v}{\partial y} = 0 \quad (1)$$

where u and v are components of velocity in x and y directions, respectively.

Momentum:

$$\rho_v \left(u_v \frac{\partial u_v}{\partial x} + v_v \frac{\partial u_v}{\partial y} \right) = -\frac{\partial p}{\partial x} + \mu_v \left(\frac{\partial^2 u_v}{\partial x^2} + \frac{\partial^2 u_v}{\partial y^2} \right) \quad (2)$$

$$\rho_v \left(u_v \frac{\partial v_v}{\partial x} + v_v \frac{\partial v_v}{\partial y} \right) = -\frac{\partial p}{\partial y} + \rho_v g + \mu_v \left(\frac{\partial^2 v_v}{\partial x^2} + \frac{\partial^2 v_v}{\partial y^2} \right) \quad (3)$$

Energy:

$$\rho_v c_p \left(u_v \frac{\partial T}{\partial x} + v_v \frac{\partial T}{\partial y} \right) = k_v \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right) \quad (4)$$

where, g is the acceleration of gravity, ρ_v vapor density, μ_v is the effective viscosity of vapor for laminar case is merely the dynamic viscosity, C_p specific heat and k_v is thermal conductivity of vapor.

Liquid wick region:

$$\text{Continuity: } \frac{\partial u_l}{\partial x} + \frac{\partial v_l}{\partial y} = 0 \quad (5)$$

where, u and v are components of velocity in x and y directions, respectively

Momentum:

$$\rho_l \left(u_l \frac{\partial u_l}{\partial x} + v_l \frac{\partial u_l}{\partial y} \right) = -\frac{\partial p_l}{\partial x} + \mu_l \left(\frac{\partial^2 u_l}{\partial x^2} + \frac{\partial^2 u_l}{\partial y^2} \right) + R_x \quad (6)$$

$$\rho_l \left(u_l \frac{\partial v_l}{\partial x} + v_l \frac{\partial v_l}{\partial y} \right) = -\frac{\partial p_l}{\partial y} + \rho_l g + \mu_l \left(\frac{\partial^2 v_l}{\partial x^2} + \frac{\partial^2 v_l}{\partial y^2} \right) + R_y \quad (7)$$

R_x and R_y are distributed resistance components in x and y directions, respectively. A distributed resistance is a convenient way to approximate the effect of porous media. Any fluid element with a distributed resistance will have a real constant set number greater than 1 (ANSYS Release 11.0 documentation 2008). The total liquid pressure gradients along x and y directions are given by (ANSYS Release 11.0 documentation 2008):

$$\frac{\partial p_l}{\partial x_{\text{resistance}}} = \{-K\rho_l|u_l| + \frac{f}{D_h}\rho_l|u_l| + C\mu_l\} \quad (8)$$

$$\frac{\partial p_l}{\partial y_{\text{resistance}}} = \{-K\rho_l|v_l| + \frac{f}{D_h}\rho_l|v_l| + C\mu_l\} \quad (9)$$

where, K is the localized head loss, ρ is the density, μ is the viscosity, f is the friction coefficient and C is the FLOTTRAN permeability expressed in m^2 , which is inverse of the physical permeability (ANSYS Release 11.0 documentation 2008).

Energy:

$$\rho_l c_{p,l} \left(u_l \frac{\partial T_l}{\partial x} + v_l \frac{\partial T_l}{\partial y} \right) = k_e \left(\frac{\partial^2 T_l}{\partial x^2} + \frac{\partial^2 T_l}{\partial y^2} \right) + Q_v \quad (10)$$

where, g , ρ , μ , C_p , k_e and Q_v are gravitational acceleration, density, dynamic viscosity, specific heat, effective thermal conductivity for liquid-wick structure and volumetric heat flux respectively. Subscripts v and l refers to vapor and liquid regions respectively. k_e is the effective thermal conductivity of the liquid-wick structure for sintered powder wick, as expressed by (Peterson 1994):

$$k_e = \frac{k_l[(2k_l+k_w)-2(1-\phi)(k_l-k_w)]}{(2k_l+k_w)+(1-\phi)(k_l-k_w)} \quad (11)$$

For screen mesh wick, k_e calculated from (Peterson 1994):

$$k_e = \frac{k_l[(k_l+k_w)-(1-\phi)(k_l-k_w)]}{(k_l+k_w)+(1-\phi)(k_l-k_w)} \quad (12)$$

where, ϕ is porosity, k_l and k_w are thermal conductivity of liquid and wick material respectively.

The steady state thermal conductivity equation to predict the wall temperature is:

$$k_s \left(\frac{\partial^2 T_s}{\partial x^2} + \frac{\partial^2 T_s}{\partial y^2} \right) = 0 \quad (13)$$

where, k_s is solid thermal conductivity and T_s is solid (wall) temperature.

Boundary conditions:

Fig. 2 illustrates the detailed boundary conditions of the computational domain.

Fig. 2. Illustration of boundary conditions

At both ends of the heat pipe $u_v = v_v = u_l = v_l = 0$, and $P_v = P_l$

At the centerline of evaporator section $v_v = 0$, $\frac{\partial u_v}{\partial y} = 0$ and $\frac{\partial \tau}{\partial y} = 0$

At the centerline of condenser section $u_v = 0$, $\frac{\partial v_v}{\partial x} = 0$ and $\frac{\partial \tau}{\partial x} = 0$

For the solid (container) – liquid (wick) interface $u_l = v_l = 0$

At the adiabatic section, $\rho_v v_v = \rho_l v_l = 0$

The continuity of mass fluxes in y direction at the vapor liquid interface yields

$\rho_v v_v = \rho_l v_l = -\rho_v v_l$ where v_l is the vapor injection velocity expressed as (Nouri-Borujerdi and Layeghi 2004):

$$v_1 = \frac{Q_h}{2\rho_v \pi R_v L_e h_{fg}} \quad (14)$$

Similarly, the continuity of mass fluxes in x direction at the vapor liquid interface yields

$\rho_v u_v = \rho_l u_l = \rho_v u_l$ where u_l is the vapor suction velocity as given by (Nouri-Borujerdi and Layeghi 2004):

$$u_1 = \frac{Q_h}{2\rho_v \pi R_v L_c h_{fg}} \quad (15)$$

Where, L_e and L_c are lengths of the evaporator and condenser respectively, and h_{fg} is the latent heat of the working fluid.

The interface temperature (T_{int}) is calculated by the Clausius-Clapeyron equation, assuming the saturation temperature (T_0) and vapor pressure (P_0) at the liquid-vapor interface (Kaya and Goldak 2007):

$$T_{int} = \frac{1}{\frac{1}{T_0} - \frac{R}{h_{fg}} \ln\left(\frac{P_v}{P_0}\right)} \quad (16)$$

For the solid-liquid interface

At the evaporator part $K_e \frac{\partial T_l}{\partial y} = k_s \frac{\partial T_s}{\partial y}$

At the condenser part $K_e \frac{\partial T_l}{\partial x} = k_s \frac{\partial T_s}{\partial x}$

$$\left\{ \begin{array}{l} \text{Evaporator } k_s \frac{\partial \tau}{\partial y} = q_e \\ \text{At the external heat pipe wall } \left\{ \begin{array}{l} \text{Adiabatic } \frac{\partial \tau}{\partial y} = 0 \text{ and } \frac{\partial \tau}{\partial x} = 0 \\ \text{Condenser } -k_s \frac{\partial \tau}{\partial x} = h(T_s - T_a) \end{array} \right. \end{array} \right.$$

where, h is convection heat transfer coefficient, and T_s and T_a are wall surface and ambient temperatures respectively.

3. MESHING AND NUMERICAL SOLUTION

The modeling approach was adapted to the works of (Zhu and Vafai 1999), (Kaya and Goldak 2007) and (Nouri-Borujerdi and Layeghi 2004), taking into account the amendments to suit the L-shape heat pipe and the simulations were done by using ANSYS-FLOTTRAN. The heat from the base was transferred to the heat pipe and then to the fins and from fins to the surrounding air. In the present two-dimensional numerical simulation of the flat heat pipe an obstacle was that the top view section was broad (7.6 mm) while the side view section was thin (3.4 mm) as shown in Fig. 1(a), and this prompted to calculate the equivalent diameter of the heat pipe. Accordingly, by assuming the flat heat pipe as cylindrical heat pipe of equivalent perimeter, the equivalent diameter of the flat heat pipe was obtained as 6 mm.

The computational domain is divided into two estranged regions: 1) vapor region, and 2) liquid-wick structure and heat pipe wall. The simulation is carried out by using the element FLUID141. The vapor region is meshed with 4900 fine triangular elements, whereas the liquid-wick and solid regions are given relatively very finer meshing, forming 52456 triangular elements. Preconditioned Conjugate Gradient (PCG) was used as iterative solvers. The convergence criterion was taken as $1e-007$ were the convergence criterion is a value below which the convergence monitors for each active degree of freedom must drop to cause the time step to terminate (ANSYS Release 11.0 documentation 2008). For solving the set of vapor matrix equations, the Conjugate Residual Method was used; however when solving equations of solid and liquid simultaneously, the non-symmetric matrix equations appeared, and to overcome this phenomenon the Preconditioned Biconjugate Gradient Stabilized Method (BiCGSTAB) was selected.

4. RESULTS AND DISCUSSION

4.1 Natural Convection

In the natural convection mode, the simulation was performed with heat input of 20 W. At natural convection, the physical properties of the working fluid were taken at 120 °C as summarized in Table 2. The wick structure consisted of sintered copper powder wick which has fine pore size causing low permeability (1.17×10^{-11} m²) (Mwaba et al. 2006). The effective thermal conductivity of liquid-wick structure is 3.69 W/mK, which was calculated from Eq. (11).

The wall temperature results were compared with experimental data to verify the accuracy of the simulation results at the same operation conditions.

Table 2 Physical properties of water and vapor at 120 °C (Faghri 1995)

	Water	Vapor
Density kg/m ³	945	1.12
Viscosity kg/m.s	2.6×10^{-4}	1.34×10^{-5}
Thermal conductivity W/mK	0.682	0.0234
Latent heat J/kg	2.2×10^6	-
Surface tension N/m	0.055	-
Pressure N/m ²	-	2.02×10^5

As shown in Fig. 3, the highest temperature reaches to 126.74 °C, located in the evaporator and then falls relatively in the adiabatic section to 120 °C and the temperature stay uniform in this region and then the temperature drops again to 114.94 °C in the condenser section. It is worth noting that the simulation temperatures including evaporator temperature (T_e), adiabatic section temperature (T_{ad}), condenser temperatures (T_{c1}) and (T_{c2}) are 125.19 °C, 120.0 °C, 119.45 °C, and 114.87 °C respectively which are close to the experimental results except in the adiabatic section temperature (T_{ad}) which is less satisfactory due to the assumption of no heat loss in the adiabatic section. Moreover, as the experiment is isolated by glass tunnel, the adiabatic temperature affected by the removal heat. The good agreement indicates the validity of the present simulation methodology.

Fig. 3. Wall temperature distribution along the heat pipe at $Q=20$ W in natural convection and comparison with the experimental results.

Methanol as working fluid

A comparative study was made by replacing water with methanol as working fluid. In the natural convection mode, the simulation was performed with heat input of 20 W. At $Q=20$ W and natural convection, the physical properties of the working fluid were taken at 120 °C as summarized in Table 3.

Table 3 Physical properties of liquid and vapor of methanol at 120 °C

	Liquid	Vapor
Density kg/m ³	696	7.7
Viscosity kg/m.s	2.0×10^{-4}	1.28×10^{-5}
Thermal conductivity W/mK	0.196	0.0022
Latent heat J/kg	0.95×10^6	-
Surface tension N/m	0.0135	-
Pressure N/m ²	-	6.41×10^5

The use of screen mesh wick with porosity 0.62 and permeability 1.93×10^{-10} (Mwaba et al. 2006) and water as working fluid resulted in value of effective thermal conductivity of water-wick structure 1.42 W/mK according to Eq. (12). As noted in Table 3, the

thermal conductivity of methanol is small (0.196 W/mK) compared with water (0.682 W/mK) at the same operating temperature. This leads to low effective thermal conductivity of Liquid-wick region 0.445 W/mK. In order to know the influence of the wick thickness on the thermal performance of L-shape heat pipe in the present simulation, the wick thicknesses were taken as 0.5 mm to 0.75 mm for using both water and methanol as working fluids.

Fig. 4 shows the wall temperature distribution of the heat pipe in natural convection with reference to change in working fluid and thicknesses of screen mesh wick ($K=1.93 \times 10^{-10}$ m²) at $Q=20$ W. It is observed that use of water as working fluid leads to small temperature difference between evaporator and condenser compared to that with methanol as working fluid. This is attributed to the fact that the thermal conductivity of water is higher than that of methanol, which leads to the increase of effective thermal conductivity of water-wick region. The effect of wick thickness is also observed in Fig. 4; the temperature difference of the heat pipe with wick thickness 0.5 mm is less than that with wick thickness 0.75 mm. Water with 0.5 mm screen mesh wick recorded the smallest temperature difference (24.56 °C) while methanol with 0.75 mm screen mesh wick recorded the largest (54 °C). It is also observed that the temperature distribution at the adiabatic region is uniform and constant at about 120 °C for all the cases.

Fig. 4. Wall temperature distribution of heat pipe in natural convection with reference to variation of working fluid and thicknesses of screen mesh wick ($K=1.93 \times 10^{-10}$ m²) at $Q=20$ W.

The comparison of heat pipes with screen mesh wick and different working fluids and wick thicknesses, is shown in Fig. 5. With increasing wick thickness, the flow area increases and it causes decrease in pressure drop in the liquid-wick region. It is observed that the maximum liquid pressure drop (692.43 Pa) is obtained for methanol with 0.5 mm wick thickness while the minimum pressure drop (338.94 Pa) is obtained for water with 0.75mm wick thickness. Although the liquid pressure drop of methanol is large owing to its higher viscosity, the pressure drop of water with wick thickness of 0.5 mm (501.81 Pa) is greater than that of methanol with wick thickness of 0.75 mm (468.37 Pa). This indicates that the effect of thickness on the pressure drop is more significant than the effect of the type of liquids used.

4.2 Forced Convection

In this mode, the applied heat is 35 W with coolant airflow rate of 6.5m³/h. As shown in Fig. 6, the highest temperature (52.94 °C) is located at the evaporator section. The adiabatic area between the evaporator and condenser has uniform temperature distribution about 47 °C, and it begins to decline at the condenser region to reach to 43.05 °C towards the end of the condenser section.

Fig. 5. Liquid pressure distribution along the heat pipe at $Q=20$ W in natural convection with various wick thicknesses (Screen mesh $K=1.93 \times 10^{-10} \text{ m}^2$).

Fig. 6 shows the wall temperature distribution along the heat pipe at coolant airflow of $6.5 \text{ m}^3/\text{h}$. The predict temperatures T_e , T_{ad} , T_{c1} and T_{c2} are observed to be 51.52°C , 47.00°C , 46.23°C , and 43.06°C respectively; this decline in temperatures is obviously due to the enhanced cooling by forced convection. These values of temperatures are in good agreement with experimental results except evaporator temperature (T_e). This is due to (T_e) being affected by cooling air flow which decreases it.

Fig. 6. Predicted temperature distribution along heat pipe at $Q=35$ W with coolant airflow rate of $6.5 \text{ m}^3/\text{h}$ and comparison with the experimental results.

In order to show the effect of parameters and compare their influences on the performance of heat pipe under forced convection, the parameters are collected in Fig. 7. It is observed that the use of sintered copper powder wick leads to small temperature difference between evaporator and condenser compared to screen mesh wick as the sintered copper wick has low permeability $1.17 \times 10^{-11} \text{ m}^2$, which causes increase of effective thermal conductivity of wick region. Additionally, the use of water as working fluid leads to small temperature difference compared to that with methanol as working fluid. This is because the thermal conductivity of water is higher than that of methanol, which increases the effective thermal conductivity of water-wick region compared to the methanol-wick region. The effect of wick thickness is also observed; the temperature difference of the heat pipe with wick thickness 0.5 mm is less than that of heat pipe with wick thickness 0.75 mm . The water with 0.5 mm sintered copper wick recorded the smallest temperature difference (9.8°C) while methanol with 0.75 mm screen

mesh wick recorded the largest temperature difference (28.27°C). It is also observed that there is a rapprochement between the temperature distribution of water with 0.75 mm screen mesh wick and methanol with 0.5 mm screen mesh wick. Additionally, it is observed that the temperature distribution at the adiabatic region is uniform and constant about 47°C for all cases. Methanol with copper powder wick was excluded because it results in exceedingly high pressure in the liquid-wick region.

Fig. 7. Wall temperature distribution along the heat pipe at $Q=35$ W and forced convection ($Q^*=6.5 \text{ m}^3/\text{h}$) with various working fluid and wick types and thicknesses.

In the forced convection, heat input of 35 W is used, with increasing heat input, the flow rate of liquid increases, so it causes of increasing the pressure drop in liquid region. Fig. 8 shows the liquid pressure distribution along the heat pipe at $Q=35 \text{ W}$ under forced convection ($Q^*=6.5 \text{ m}^3/\text{h}$) with various sintered copper wick thicknesses. It is observed that liquid pressure drop is decreased from 6563 Pa to 4219 Pa when the wick thickness is increased from 0.5 mm to 0.75 mm , as increasing the thickness of the wick leads to increase in the liquid flow area through the wick, which causes reduction in liquid pressure drop liquid-wick region.

Fig. 8. Predict liquid (Water) pressure distribution at $Q=35 \text{ W}$ in forced convection ($Q^*=6.5 \text{ m}^3/\text{h}$) of sintered copper powder wick with various thicknesses.

Fig. 9 clearly shows the comparison of heat pipes with screen mesh wicks of difference working fluids and wick thicknesses. It is observed that the maximum liquid pressure drop (1789 Pa) is obtained when methanol with 0.5 mm wick thickness is used, while the minimum pressure drop (633 Pa) is obtained when water with 0.75 mm wick thickness is used. Although the liquid pressure drop of methanol is

large due to its higher viscosity, the pressure drop of methanol with wick thickness of 0.75 mm (1083 Pa) is less than the pressure of water with wick thickness of 0.5 mm (1129 Pa). This is because, the increased thickness of the wick leads to increase in the liquid flow area through the wick, which causes reduction in liquid pressure drop at the liquid-wick region.

Fig. 9 Liquid pressure distribution along the heat pipe at $Q=35W$ at $Q=6.5 \text{ m}^3/\text{h}$ with various working fluid and wick thickness at Screen mesh

5. CONCLUSION

A two-dimensional FEM simulation of working fluid behavior inside L-shape heat pipe was performed by using ANSYS-FLOTTRAN software to study the effect of thickness and permeability of wick structure using water and methanol as different working fluids on L-shape heat pipe performance.

The study concluded with the following key points:

- The increase in thickness of the wick caused a decrease in liquid pressure drop in the liquid-wick region while the use of methanol as working fluid increased the liquid pressure.
- The use of sintered copper powder wick resulted in small temperature difference between the evaporator and condenser compared to screen mesh wick.
- Although good performance of heat pipe was achieved by employing water with 0.5 mm thickness of sintered copper powder, the unfavorable high pressure resulted in low heat transport capability at the liquid-wick region.

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