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Chapter 3

Heat Pipe and Phase Change Heat Transfer Technologies for Electronics Cooling

Chan Byon

Additional information is available at the end of the chapter

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Abstract

The heat pipe is a well-known cooling module for advanced electronic devices. The heat pipe has many applications, particularly in electronics and related area such as PC, laptop, display, artificial satellite, and telecommunication modules. The heat pipe utilizes phase change heat transfer inside enveloped structures, where the working fluid evaporates in heated zone, and vapor moves to the condenser, and the condensed liquid is pumped back through microporous structure call wick. The performance of applicability in electronics of heat pipe is strongly dependent on the geometry, working fluid, and microstructure of wick. Therefore, it is worth considering the theory and technologies related to heat pipes for advanced electronics cooling. According to the purpose of this chapter mentioned above, the author considers fundamental aspects regarding heat pipe and phase change phenomena. First, the working principle of heat pipe is introduced. Important parameters in heat pipe are considered, and theoretical model for predicting the thermal performance of the heat pipe is introduced. In addition, design method for heat pipe is presented. Finally, applications of heat pipe to electronics cooling are presented. This chapter covers knowledge and state-of-art technologies in regard to heat pipe and phase change heat transfer. For a reliable operation of future electronics that have ultra-high heat flux amounts to 1000 W/m^2, heat pipe and phase change heat transfer are essential. This chapter provides the most valuable opportunity for all readers from industry and academia to share the professional knowledge and to promote their ability in practical applications.

Keywords: Heat pipe, phase change, wick, design, analysis

1. Introduction

Effective cooling technology is a crucial requirement for a reliable operation of electronic components. The electronics cooling methods can be hierarchically classified as chip level

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cooling, package level cooling, and system level cooling, depending on the geometrical scale. In the package or system level cooling, the cooling modules such as heat sinks and heat pipes are widely employed for an efficient dissipation of heat as well as uniform temperature distribution. Especially, the use of heat pipes for electronics cooling has recently been increasing abruptly because the heat pipe is an attractive passive cooling scheme, which can offer high effective thermal conductivity and large heat transport capability. As shown in **Figure 1**, the heat pipes have been conventionally used for PCs, laptops, telecommunication units, solar collectors, small energy systems such as geothermal pipes, and satellites. Recently, the application of heat pipe even includes smart phones, vehicle headlight, gas burner, LED products, and agricultural systems, as shown in **Figure 2**.



Figure 1. Heat pipe applications.

The heat pipe is a thermal superconductor of which thermal conductivity amounts to several thousands of Watts per meter-Kelvin. Due to the extremely high effective thermal conductivity, the heat pipe can handle a large amount of heat transfer with a negligible temperature drop. In addition, the heat pipe is a passive cooling module, which accompanies no power consumption or moving parts. Literally, the heat pipe is apparently just a pipe without any accessories for operating it. Furthermore, the shape of the heat pipe does not necessarily have to be cylindrical, but it can be formed into various shapes such as disks, flat plates, and airfoils. Attributable to these characteristics, the heat pipe is regarded as an ultimate candidate for addressing the thermal problem of concurrent high-power-density semiconductor industry, which encompasses solar cell, LEDs, power amplifiers, lasers, as well as electronic devices.

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Figure 2. Recent applications.

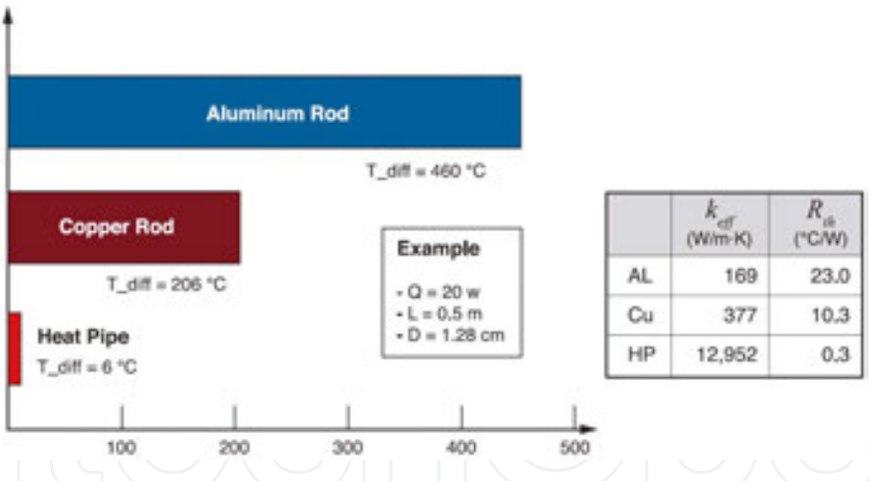


Figure 3. The superiority of heat pipe over other thermally conducting materials.

Figure 3 clearly illustrates the superiority of the heat pipe. The goodness of the heat transfer module is characterized by the effective thermal conductivity (k_{eff}) or thermal resistance (R_{th}) of the module. As an example, a typical value of effective thermal conductivity of a copper–water heat pipe with 0.5-m length and 1/2 inch diameter is around 10,000 W/mK, which is much larger than those of thermally conductive metals such as copper (~377 W/mK) or aluminum (~169 W/mK). This results in very low thermal resistance (~0.3 K/W), indicating low temperature drop with respect to the given thermal load. When 20 W heat is applied, this heat pipe would yield 6°C temperature difference between heat source and sink, whereas metal

rods with same geometry have 206°C and 460°C for copper and aluminum, respectively. Provided that the ambient air is at 20°C, the chip temperature is only 26°C, which enables designers to easily come up with plausible and fascinating thermal solution.

In this chapter, general aspects of heat pipes for electronics cooling are introduced. The contents cover the working principle of heat pipes, design and analyzation methods, components and structure of heat pipes, implementation in electronics cooling, characterization and theories, and design and manufacturing process.

2. Working principle

2.1. Introduction to working principle

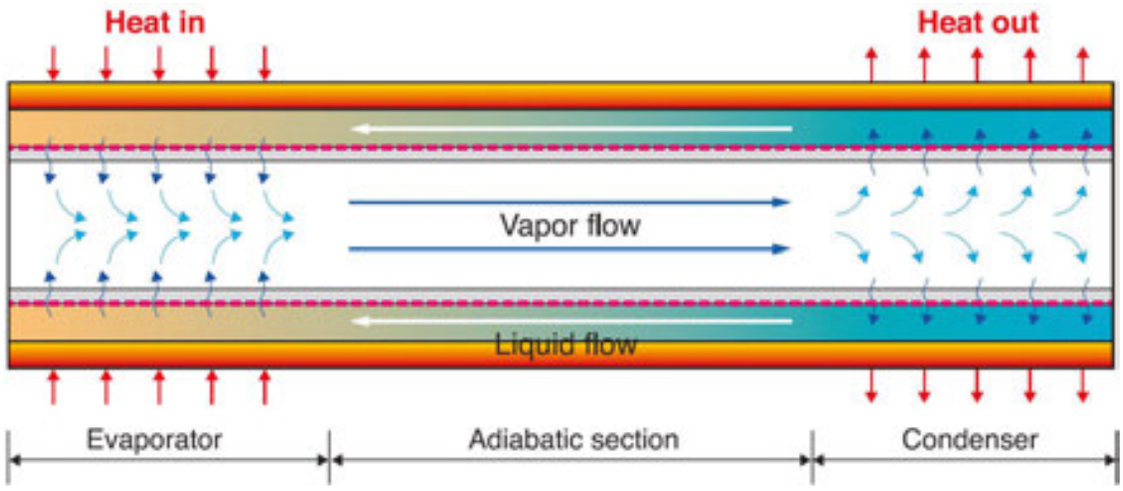


Figure 4. Working principle of heat pipe.

The working principle of the heat pipe is summarized in **Figure 4**. The heat pipe consists of metal envelope, wick, and working fluid. The wick is a microporous structure made of metal and is attached to the inner surface of the envelope. The working fluid is located in the void space inside the wick. When the heat is applied at the evaporator by an external heat source, the applied heat vaporizes the working fluid in the heat pipe. The generated vapor of working fluid elevates the pressure and results in pressure difference along the axial direction. The pressure difference drives the vapor from evaporator to the condenser, where it condenses releasing the latent heat of vaporization to the heat sink. In the meantime, depletion of liquid by evaporation at the evaporator causes the liquid–vapor interface to enter into the wick surface, and thus a capillary pressure is developed there. This capillary pressure pumps the condensed liquid back to the evaporator for re-evaporation of working fluid. Likewise, the working fluid circulates in a closed loop inside the envelope, while evaporation and condensation simultaneously take place for heat absorption and dissipation, respectively. The high thermal performance of the heat pipe is originated from the latent heat of vaporization, which typically amounts to millions of Joules per 1 kg of fluid.

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2.2. Wick for heat pipe

The flow in the wick is attributable to the same mechanism with the suction of water by a sponge. The microsized pores in the sponge (or wick) can properly generate the meniscus at the liquid–vapor interfaces, and this yields the capillary pressure gradient and resulting liquid movement. It should be noted that the wick provides the capillary pumping of working fluid, which must be steadily supplied for the operation of heat pipe as well as the flow passage of the working fluid. In addition, the wick also acts as a thermal flow path because the applied heat is transferred to the working fluid through the envelope and wick. Therefore, the thermal performance of the heat pipe is strongly dependent on the wick structure.

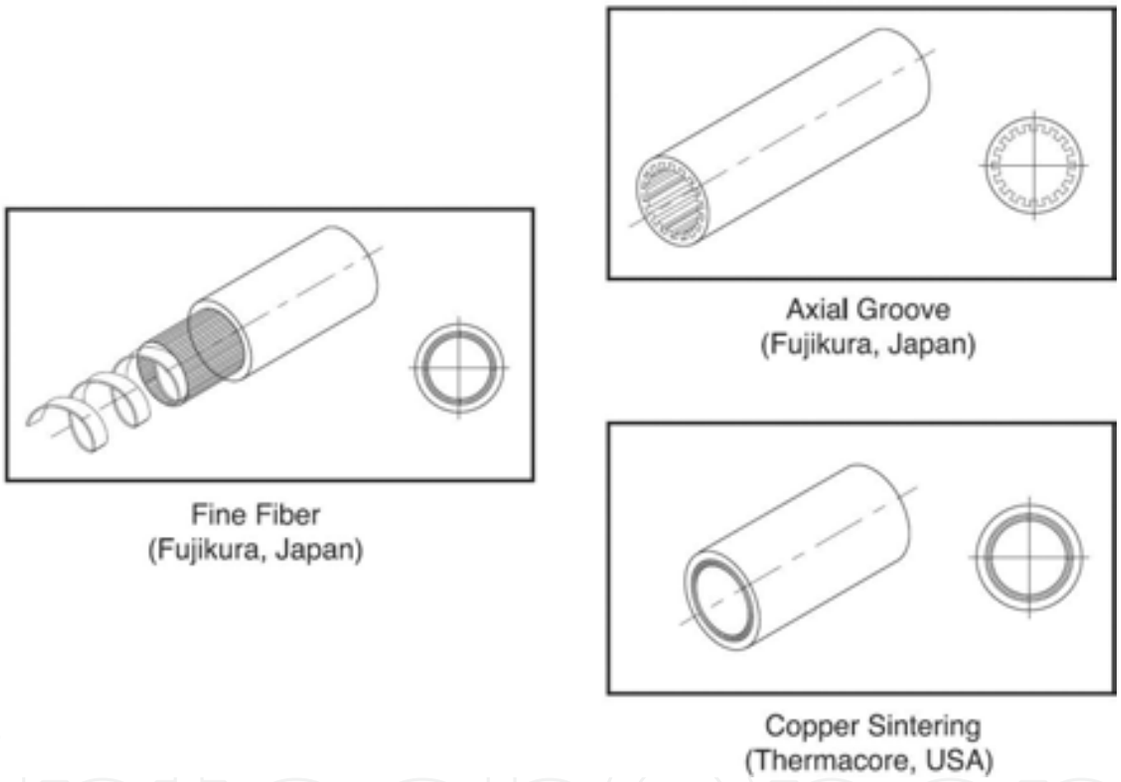


Figure 5. Typical wick structures.

In this regard, various types of wick structures have been used for enhancing the thermal performance of heat pipes. **Figure 5** shows three representative types of wick structures: mesh screen wick (it is also often termed as fiber mesh or wrapped screen), grooved wick, and sintered particle (or sintered powder) wick. The mesh screen wick is the most common wick structure, which made of wrapped textiles of metal wires. The grooved wick utilizes axial grooves directly sculptured on the envelope inner surface as the flow channel. The sintered particle wick is made of slightly fusing microsized metal particles together in the sintering process. The major characteristics of aforementioned wick types are shown in **Figure 6**. The mesh screen wick can have high capillary pressure and moderate permeability because numerous pores per unit length and the tightness of the structure can be controlled, where the

permeability is a measure of the ability of a porous medium to transmit fluids through itself under a given pressure drop as follows:

$$U = -\frac{K}{\mu}\frac{dP}{dx} \tag{1}$$

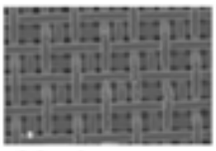
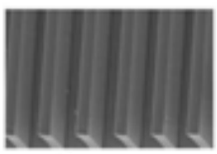
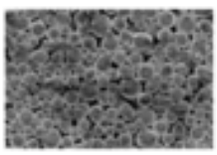
Wick structure	Capillary force	Permeability	Thermal conductivity
 Mesh screen	High	Low-average	Low
 Groove	Low	Average-high	High
 Sintered particle	High	Low-average	Average-high

Figure 6. Characteristics of wick structures.

where K is the permeability, U is the mean velocity of flow inside porous media, μ is the viscosity, and dP/dx is the applied pressure gradient.

However, the effective thermal conductivity is low because the screens are not thermally connected to each other. In case of grooved wick, the effective thermal conductivity is high due to the sturdy thermal path. It has an additional advantage in which the wide and straight (not tortuous) flow path can bring about high permeability. However, the capillary pressure is strongly limited, due to the fact that the scale of the grooves, which are machined through extrusion process, cannot be reduced beyond several tens of micrometers. It should be noted that the maximum developable capillary pressure is inversely proportional to the characteristic length of the pore structure. On the other hand, the sintered particle wick has high capillary pressure as well as moderate-to-high effective thermal conductivity, due to the tailorable particle size and fused contact between particles. However, the permeability of the sintered particle wick is relatively low, due to the narrow and tortuous flow path. As shown, the type of wick has its pros and cons. Therefore, the designers choose the wick type in accordance with the corresponding suitable applications.

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3. Thermal performance

3.1. Various mechanisms

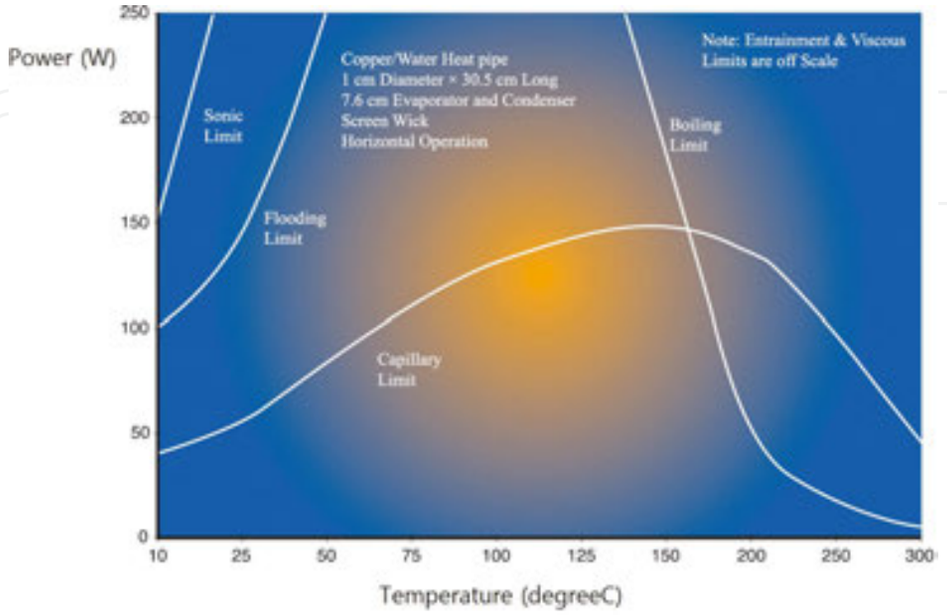


Figure 7. Performance limitations with respect to the temperature.

In case of other cooling modules, there is no heat transfer 'limitation,' implying that increasing heat transfer rate just keeps increasing the temperature drop and worsening the situation. On the contrary, there is a definite limitation of thermal performance for heat pipe, beyond which the heat transfer rate cannot be increased for a reliable operation. The thermal performance of the heat pipe is limited by one of various mechanisms depending on the working temperature range and geometry of the heat pipe. The viscous limit typically occurs during unsteady start-up at low temperature, when the internal pressure drop is not large enough to move the vapor along the heat pipe. The sonic limit also typically takes place during unsteady start-up at low temperature, when the chocked flow regime is reached at the sonic speed of the vapor. The capillary limit is related to the ability of the wick to move the liquid through the required pressure drop. It happens when the circulation rate of working fluid increases so that the pressure drop along the entire flow path reaches the developed capillary pressure. When the capillary limit happens, dry out occurs in the evaporator, while more fluid is vaporized than that can be supplied by the capillary action of the wick. The entrainment limit is related to the liquid–vapor interface where counterflows of two phases are met. In some circumstances, the drag imposed by the vapor on the returning liquid can be large enough to entrain the flow of condensate in the wick structures, resulting in dry out. The boiling limit is known to occur when the bubble nucleation is initiated in the evaporator section. The bubble generally cannot easily escape the wick with microsized pores and effectively prevent the liquid to wet the heated surface, which in turn results in burnout. The burnout heat flux is known to typically range from 20 to 30 W/cm^2 for sintered particle wicks.

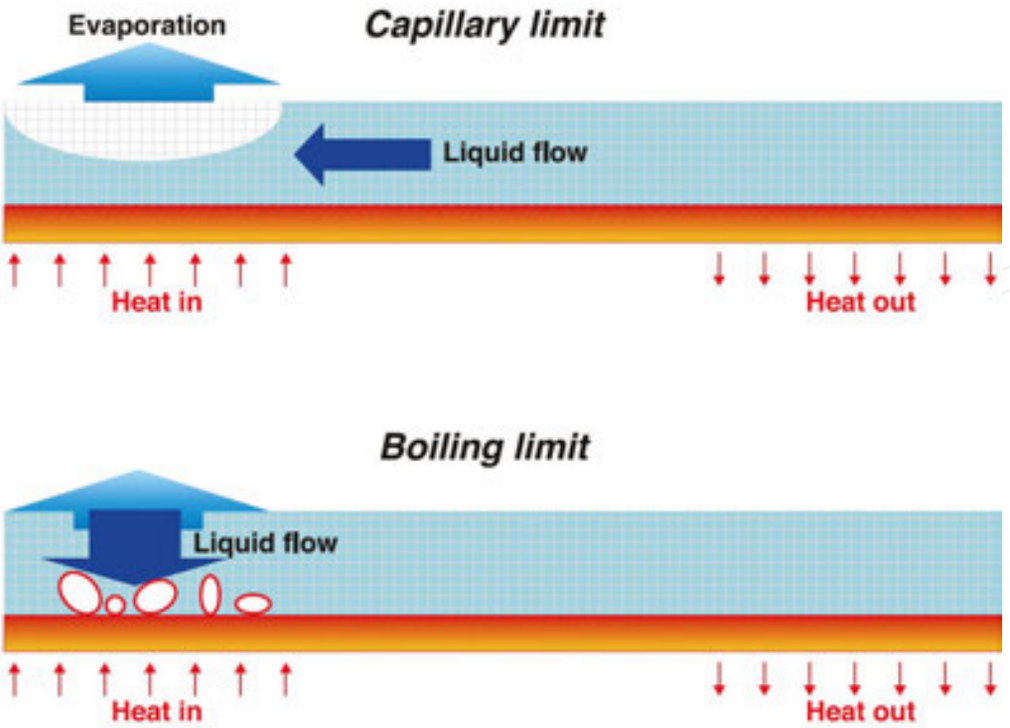


Figure 8. Capillary limit and boiling limit.

Figure 7 shows the thermal capacity of a heat pipe (copper–water, 1 cm diameter, 30 cm long) determined by various limiting mechanisms with respect to temperature. As shown in this figure, the viscous limit, the sonic limit, and the entrainment limit do not play an important role in determining the thermal capacity of the heat pipe, unless the temperature is very low (< −20°C). The heat pipes operating above atmospheric temperature are practically only governed by the capillary limit or the boiling limit, as shown in **Figure 7**. There is a good method for distinguishing those two limitations (see **Figure 8**). The capillary limit occurs when the liquid flow along the axial direction cannot afford the evaporation rate due to the limited capillary pressure. The boiling limit occurs when the bubble barricades the liquid flow onto the heated surface. In other words, the capillary limit is represented by the axial (or lateral) fluid transportation limit while the boiling limit is represented by the radial (or vertical) fluid transportation limit. It should be noted that, in the heat pipe, the limitation on the fluid transport represents the heat transfer limit because the heat transfer rate is given as the multiplication of latent heat coefficient and mass flow rate of working fluid. In this regard, the boiling limit becomes dominant when the effective heat pipe length is relatively small, and vice versa. The boiling limit also becomes important when the operating temperature is high because bubble nucleation is more likely to happen in high superheat. The next two subsections will be devoted to the models for capillary limit and boiling limit, respectively.

3.2. Capillary limit

The capillary limit is also called the wicking limit. As mentioned, the capillary limit occurs when the liquid flow along the axial direction cannot afford the evaporation rate. This situation

happens under the condition where the pressure drop along the entire flow path is equal to the developed capillary pressure. The pressure drop of working fluid consists of that of liquid flow path (ΔP_l), that of vapor flow path (ΔP_v), additional pressure drop imposed by counter-flow at the phase interface (ΔP_{l-v}), and the gravitational pressure drop (ΔP_g). Thus, the condition for the capillary limit is described by the following equation:

$$\Delta P_c = \Delta P_l + \Delta P_{l-v} + \Delta P_v + \Delta P_g \tag{2}$$

where ΔP_c is the capillary pressure difference between evaporator and condenser sections. Generally, the vapor pressure drop (ΔP_v) and interfacial pressure drop (ΔP_{l-v}) are negligible when compared with others; thus, the equation reduces to the following:

$$\frac{2\sigma}{R_{\text{eff}}} = \frac{\mu_l L_{\text{eff}}}{KA_{\text{w}}\rho_l}\dot{m} + \rho_l g L_{\text{eff}} \sin\phi \tag{3}$$

where the left-hand side represents ΔP_c, the first term on the right-hand side is ΔP_l, and the second term corresponds to ΔP_g. In this equation, σ is the surface tension coefficient, R_{eff} is the effective pore radius of the wick structure, μ_l is the liquid viscosity of working fluid, L_{eff} is the effective length of heat pipe, K is the permeability, A_w is the cross-sectional area of the wick, ρ_l is the liquid density of working fluid, $\dot{m}$ is the mass flow rate, g is the gravitational constant, and φ is the orientation angle with respect to the horizontal plane. The heat transport capacity of the heat pipe is directly proportional to the mass flow rate of the working fluid as follows:

$$Q_{\max} = h_{\text{fg}}\dot{m} \tag{4}$$

where h_{fg} is the latent heat coefficient of working fluid. Combining Equations (3) and (4) yields the following equation for the capillary limit:

$$Q_{\max} = \frac{KA_{\text{w}}h_{\text{fg}}\rho_l}{\mu_l L_{\text{eff}}}\left[\frac{2\sigma}{R_{\text{eff}}} - \rho g L_{\text{eff}} \sin\phi\right] \tag{5}$$

It should be underlined that the K and R_{eff} are related to the microstructure of the wick; h_{fg}, σ, μ_l, and ρ_l are the fluid properties; and L_{eff} and A_w represents the macroscopic geometry of the heat pipe. When the gravitational force can be neglected, the Equation (5) can be rewritten and each kind of parameters can be detached as independent term as follows:

$$Q_{\max} = \left(\frac{2\sigma h_{\text{fg}}\rho_l}{\mu_l}\right)\left(\frac{A_{\text{w}}}{L_{\text{eff}}}\right)\left(\frac{K}{R_{\text{eff}}}\right) \tag{6}$$

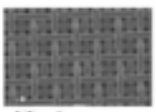
Wick type	K	R_{eff}	Typical value of K/R_{eff} (micron)
 Groove	$\frac{2\varepsilon r_{h,l}^{2}}{f\,\mathrm{Re}}$	w	~1
 Sintered particle	$\frac{\varepsilon^3 d^2}{150(1-\varepsilon)^2}$	$\frac{d}{5}$	~ 0.3
 Mesh screen	$\frac{\varepsilon^3 d^2}{122(1-\varepsilon)^2}$	$\frac{w+d}{2}$	~ 0.3

Figure 9. K and R_{eff} values for typical wick structures.

The first paragraphed term is a combination of the fluid properties, suggesting that the capillary limit of heat pipe is proportional to this term. This term is called the figure of merit of working fluid. The second paragraphed term is about the macroscopic geometry of the heat pipe. The last term is related to the wick microstructure, thus, in regard to the wick design, we have to maximize this term. This term is often called the capillary performance of wick. The permeability K is proportional to the pore characteristic length, whereas R_{eff} is inversely proportional to the pore size. Therefore, the ratio between K and R_{eff} captures a trade off between those two competing effects. The K and R_{eff} values for representative wick structure are shown in **Figure 9**.

3.3. Boiling limit

Regarding the boiling limit, it has been postulated that the boiling limit occurs as soon as the bubble nucleation is initiated. Onset of nucleate boiling within the wick was considered as a mechanism of failure and was avoided. On the basis of that postulation, the following correlation for predicting boiling limit has been widely used [1]:

$$Q_{\max} = \frac{2\pi L_e k_e T_v}{h_{\text{fg}} \rho_v \ln\left(r_i / r_v\right)} \left(\frac{2\sigma}{r_b} - P_c \right) \tag{7}$$

where L_e is the evaporator length, k_e is the effective thermal conductivity of wick, T_v is the vapor core temperature, h_{fg} is the latent heat, ρ_v is the vapor density, r_v is the vapor core radius, r_i is the radius of outer circle including the wick thickness, and σ is the surface tension coefficient. In Equation (7), important design parameters related to the wick microstructure are r_b and P_c, which are bubble radius and capillary pressure, respectively. Even though Equation (7) is simple and in a closed form, it is difficult to implement this equation in which these parameters

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are quite arbitrary, and thus, it is difficult to exactly predict those values. To accurately determine r_b and P_c, additional experiment should be performed [1]. Another fundamental problem also exists in which the nucleate boiling within the wick does not necessarily represent a heat transfer limit unless bubbles cannot escape from the wick, as indicated by several researchers [2]. Indeed, nucleate boiling may not stop or retard the capillary-driven flow in porous media according to the literatures. Some researchers even insisted that the nucleate boiling in the moderate temperature heat pipe wicks is not only tolerable but could also produce performance enhancement by significantly increasing the heat transfer coefficient over the conduction model and consequently reducing the wick temperature drop [3]. Therefore, new light should be shed on the model for the boiling limit. As illustrated in Equation (6), key parameters for capillary limit are K and R_{eff}. Equation (7) shows that key parameter for boiling limit is k_e, excluding the effect of permeability. Recently, it has been shown that the boiling limit does not occur with the nucleate boiling if the vapor bubble can escape the wick efficiently [4]. This suggests that the K is also an important parameter for the boiling limit.

4. Heat pipe designs

4.1. Heat pipe design procedure

The design procedure of the heat pipe is as follows:

1. working fluid selection,
2. wick type selection,
3. container material selection,
4. determining diameter,
5. determining thickness,
6. wick design, and
7. heat sink and source interface design.

The followed subsections will be devoted to each procedure.

4.2. Working fluid selection

The first step for designing the heat pipe is to select the working fluid according to the operating temperature of the heat pipe. Each fluid has its vapor pressure profile with respect to the temperature. The vapor pressure increases as the temperature increases, and when the vapor pressure reaches the pressure of environment, boiling occurs. The heat pipe is designed to operate nearly at the boiling temperature for facilitating the heat transfer rate associated with the latent heat. Therefore, the working fluid should be selected under the consideration of the operating temperature of heat pipe. Various kinds of working fluids and their operating

temperature ranges and corresponding inner pressures are shown in **Figure 10**. In case of water-based heat pipe that operates at the room temperature, the inner pressure of heat pipe is typically set to be approximately 0.03 bar for maximizing the thermal performance. When the operating temperature is 200°C, the inner pressure of the heat pipe should be set at roughly 16 bar. For cryogenic applications, helium or nitrogen gas is used. For medium or high temperature applications, liquid metals such as sodium and mercury are typically used. The inner pressure of heat pipe should be properly adjusted according to its operating temperature.

Range	Name	Temp(°C)	Pressure(bar)
Cryogenic	Helium	-271:-268	0.06:2.29
	Nitrogen	-203:-150	0.48:28.8
Low	Ammonia	-60:120	0.27:90.44
	Water	-20:200	0.02:16.19
	Ethanol	-30:130	0.01:4.30
	Acetone	-40:140	0.01:10.49
Medium	Naphthalene	100:400	0.02:17.5
	Mercury	150:750	0.01:63
High	Sodium	500:1300	0.01:15.91
	Caesium	375:825	0.02:3.41
	Lithium	1030:1730	0.07:8.90

Figure 10. Operating temperature of working fluids.

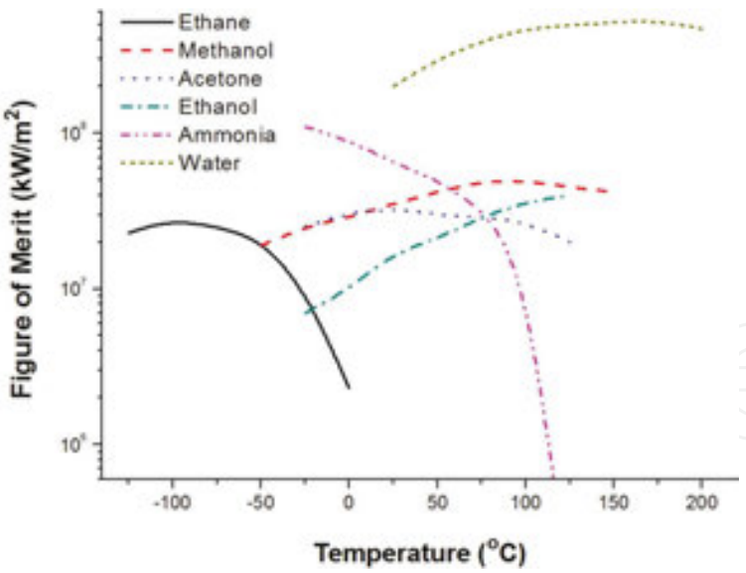


Figure 11. Figure of merit numbers of working fluids.

The working fluid selection is also important in terms of the thermal performance. Equation (6) shows that the thermal performance of heat pipe is directly proportional to a fluid property, $\rho_l \sigma h_{fg}/\mu_l$. This is often called the figure of merit of working fluid. **Figure 11** shows the figure of merit with respect to the temperature for various working fluids. As shown in this figure,

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occurring in low to moderate temperatures, water is the liquid with the highest figure of merit number. This is why the water is the most commonly used for heat pipe. Another common fluid is ammonia, which is used for low-temperature applications.

	Water	Acetone	Ammonia	Methanol
Copper	R	R	I	R
Aluminum	N	R	R	I
Stainless steel	N	I	R	N
Nickel	I	I	R	R
R	Recommended			
I	Indefinite			
N	Not compatible			

Figure 12. Material compatibility.

4.3. Wick type selection

The second step is to select the wick type. Typically, five selections can be considered: no wick (for thermosiphon), mesh screen wick, grooved wick, sintered particle wick, and heterogeneous type wick. The reason we select the wick type prior to choosing the material is that the manufacturable microstructure is dependent on the material.

4.4. Container and wick material selection

After choosing the wick type, the material for container and wick is selected. Here, the major consideration is the compatibility between the working fluid and the material. Water–copper combination is known to have a good compatibility. On the other hand, the water is not compatible with aluminum due to unpreferred gas generation. The material compatibility with working fluid is shown in **Figure 12**. Copper is shown to be compatible with water, acetone, and methanol. The aluminum has good compatibility with acetone and ammonia, but not with water.

4.5. Determination of diameter

The next step is to determine the diameter of the heat pipe. The diameter becomes a major geometric parameter upon consideration of the vapor velocity. When the diameter of heat pipe is too small, the vapor velocity increases much, and compressibility effect appears, which in turn aggravates the performance of heat pipe significantly. Typically, it is known that the compressibility effect is negligible when the Mach number is less than 0.2. To fulfill this criterion, the following equation should be satisfied.

$$d_v > \sqrt{\frac{20Q_{\max}}{\pi\rho_v h_{fg}\sqrt{\gamma_v R_v T_v}}} \tag{8}$$

where d_v is the vapor core diameter, $Q_{\max}$ is the maximum axial heat flux, ρ_v is the vapor density, γ_v is the vapor-specific heat ratio, h_{fg} is the latent heat of vaporization, R_v is the gas constant for vapor, and T_v is the vapor temperature.

4.6. Determination of thickness

As the heat pipe is like a pressure vessel, it must satisfy the ASME vessel codes. Typically, the maximum allowable stress at any given temperature can only be one-fourth of the material's maximum tensile strength. The maximum hoop stress in the heat pipe wall is given as follows [1]:

$$f_{\max} = \frac{Pd_o}{2t} \tag{9}$$

where $f_{\max}$ is the maximum stress in the heat pipe wall; P is the pressure differential across the wall, which causes the stress; d_o is the heat pipe outer wall; and t is the wall thickness. The safety criterion is given as follows:

$$f_{\max} < \frac{\sigma_Y}{4} \tag{10}$$

where σ_Y is the yielding stress of the container material. Combining Equations (9) and (10) yields:

$$\frac{2Pd_o}{\sigma_Y} < t \tag{11}$$

4.7. Wick design

The maximum thermal performance of heat pipe is given in Equation (6). Let us retrieve Equation (6) as Equation (12).

$$Q_{\max} = \left(\frac{2\sigma h_{\text{fg}}\rho_l}{\mu_l}\right)\left(\frac{A_{\text{w}}}{L_{\text{eff}}}\right)\left(\frac{K}{R_{\text{eff}}}\right) \tag{12}$$

In Equation (12), design parameters related to the wick are K and R_{eff}. The K is known to proportional to the square of the characteristic pore size, whereas R_{eff} is inversely proportional to the characteristic pore size. Therefore, the capillary performance, K/R_{eff}, is directly proportional to the characteristic pore size. However, when the pore size is too large, the capillary pressure becomes too small so that the gravity effect cannot be overcome, which in turn makes the heat pipe useless. In addition, large pore size represents significant effect of inertia force.

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It should be noted that Equation (12) is derived under the postulation that the flow rate of working fluid is determined upon the balance between capillary force and viscous friction force where the inertia force is negligible in microscale flow. When the inertia force becomes significant, the thermal performance is significantly deviated from the prediction by Equation (12), in other words, is degraded much. For these reasons, the particle size of the sintered particle wick typically ranges from 40 μm to 300 μm. In case of looped heat pipe (LHP) where extremely high capillary pressure is required, nickel particles with 1–5 μm diameter are used.

4.8. Heat sink–source interface design

Besides design of heat pipe itself, the interfaces of heat pipe with heat sink–source are also of significant interest because the interfacial contact thermal resistance is much larger than that of heat pipe itself. The contact thermal resistance between the evaporator and the heat source and that between the condenser and the heat sink is relatively large. Therefore, they have to be carefully considered and minimized.

4.9. Thermal resistance considerations

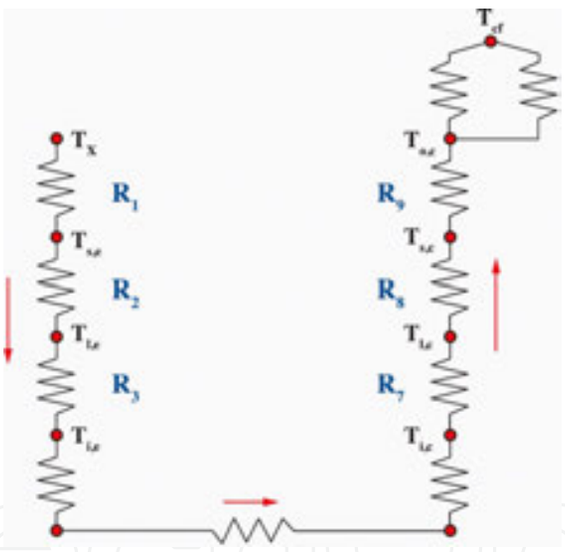


Figure 13. Thermal resistance network.

Through Sections 4.1–4.7, only the maximum heat transport capability has been regarded as the performance index of the heat pipe. However, sometimes another performance index, the thermal resistance, is more important when the heat transfer rate is not of an important consideration while the temperature uniformization is more important. The thermal resistance of the heat pipe can be estimated based on the thermal resistance network, as shown in **Figure 13**. T_x is the heat source temperature, and T_{cf} is the heat sink temperature. The subscripts *e* and *c* represent the evaporator and condenser, respectively. The subscripts *s*, *l*, and *i* represent the shell, liquid, and interface, respectively. The various thermal resistance components and correlations for predicting them are shown in **Figure 14**.

Location	Heat flow	Thermal resistance (R)
Evaporator B.C. to shell	$q_1 = \frac{1}{R_1}(T_X - T_{s,e})$	$\frac{1}{2\pi r_o L_e}[\frac{1}{h_{xx}} + \frac{\Delta r_s}{2k_s}]$
Evaporator shell to liquid	$q_2 = \frac{1}{R_2}(T_{s,e} - T_{l,e})$	$\frac{1}{2\pi r_i L_e}[\frac{\Delta r_s}{2k_s} + \frac{\Delta r_l}{2k_l}]$
Evaporator liquid to evaporator interface	$q_3 = \frac{1}{R_3}(T_{l,e} - T_{i,e})$	$\frac{1}{2\pi r_w L_e}[\frac{\Delta r_l}{2k_l}]$
Liquid-vapor interface in the evaporator	$q_4 = m'_w h_{fg} = f(T_{i,e}, T_v)$	
Vapor region	$q_5 = q_4$	
Liquid-vapor interface in the condenser	$q_6 = m'_w h_{fg} = f(T_{i,c}, T_v)$	
Liquid-vapor interface to liquid,condenser	$q_7 = \frac{1}{R_7}(T_{i,e} - T_{l,c})$	$\frac{1}{2\pi r_w L_c}[\frac{\Delta r_l}{2k_l}]$
Liquid to shell, condenser	$q_8 = \frac{1}{R_8}(T_{l,c} - T_{s,c})$	$\frac{1}{2\pi r_i L_c}[\frac{\Delta r_s}{2k_s} + \frac{\Delta r_l}{2k_l}]$
Shell to B.C., condenser	$q_9 = \frac{1}{R_9}(T_{s,c} - T_{c,f})$	$\frac{1}{2\pi r_o L_c}[\frac{1}{h_{cf}} + \frac{\Delta r_s}{2k_s}]$

Figure 14. Thermal resistance correlations.

5. Application to electronics cooling

The types of heat pipe applications to electronics cooling are as follows: use of flat-plate heat pipe, heat pipe–embedded heat spreader, block to fin, block to block, and fin to fin. The tubular heat pipe cannot solely used because its interface cannot be fully attached to the electronic devices having flat interface. For heat pipe to be adapted to the electronics cooling, the heat pipe itself should be formed into flat-plate type, or the tubular heat pipe should be fitted to rectangular-shaped based block, as shown in **Figure 15**. The heat pipe–embedded heat spreader is shown in **Figure 16**.

The block-to-fin applications are shown in **Figure 17**. The heat pipe has to be anyhow connected to the heat sink for the final heat dissipation to the air. The heat pipe–embedded block can be directly connected to the fin, as shown in this figure. In some applications such as sever computer and telecommunication unit handle a large amount of data, block-to-block module is employed, as shown in **Figure 18**. In some applications, the fin-to-fin module is also used.

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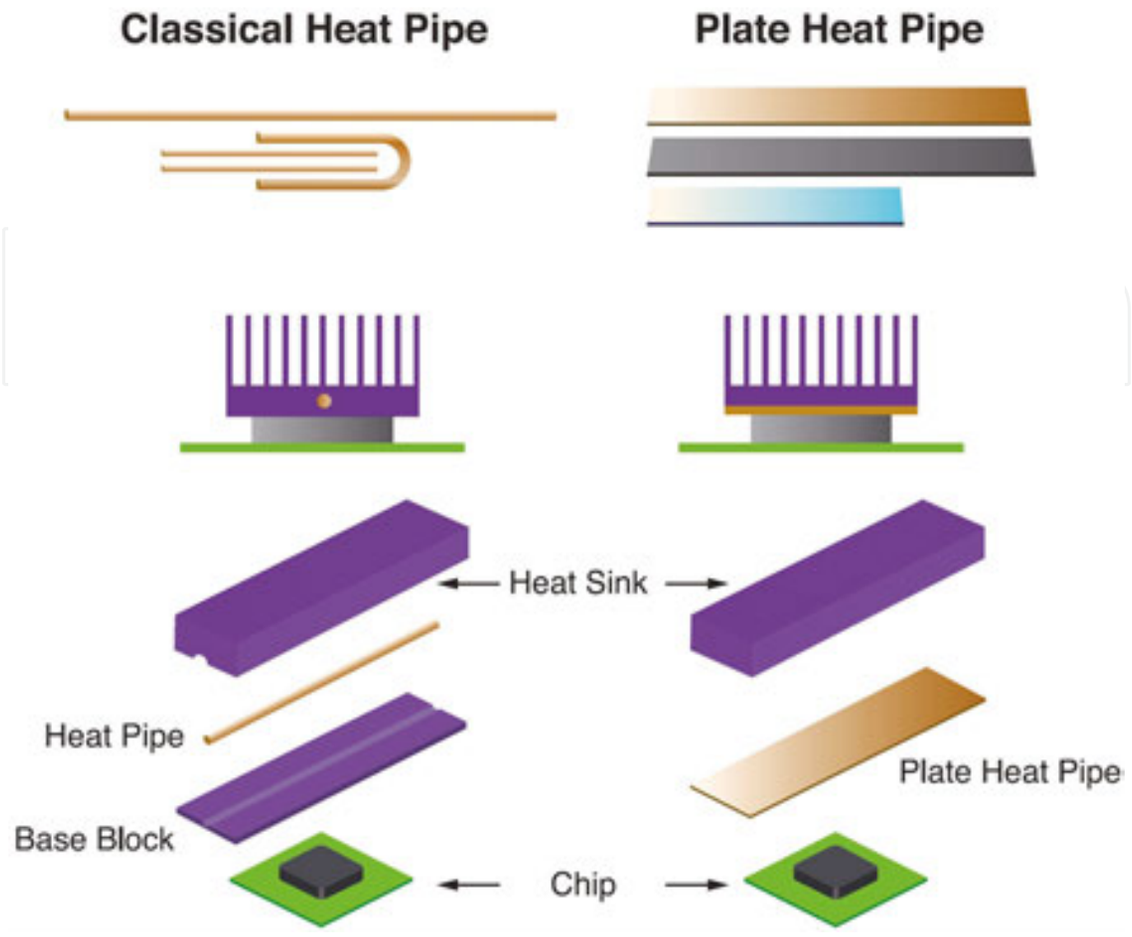


Figure 15. Use of tubular heat pipe and flat-plate heat pipe.

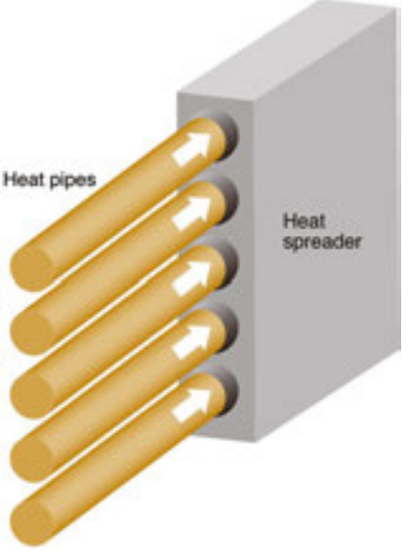


Figure 16. Heat pipe–embedded heat spreaders.

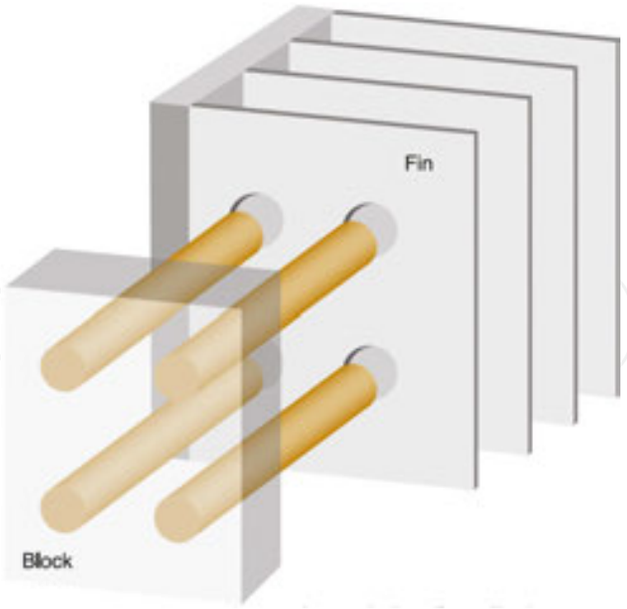


Figure 17. Block-to-fin applications.

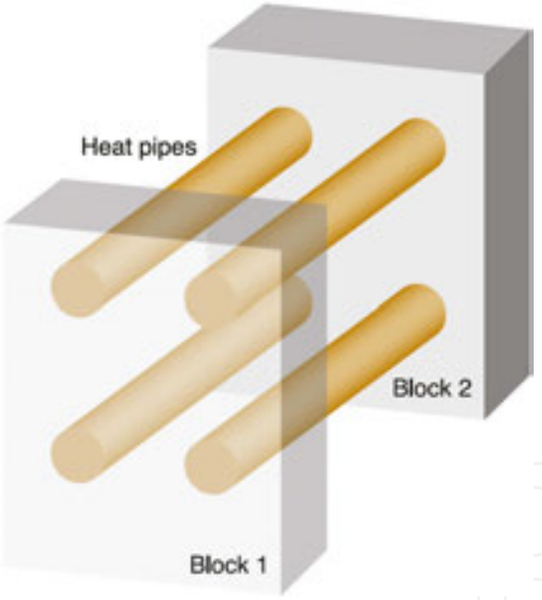


Figure 18. Block-to-block applications.

The use of heat pipe to electronics cooling is diversified into portable devices, VGA, mobile PC, LED projector and related devices, telecommunication repeater, and so on. The heat pipe is also widely employed in solar heat collection, snow melting, heat exchanger and related energy applications, and pure science applications demanding ultra-precise temperature control. Especially for the semiconductor devices whose performance and lifetime are sensitive to the temperature, heat pipe is an ultimate thermal solution. The use of heat pipe will surely expand, and it will gradually have more ripple effect in various industrial areas.

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6. Summary

In this chapter, the general aspects of heat pipes are introduced. The working principle of the heat pipe is based on two phase flows pumped by capillary pressure formed at the wick. The wick plays an important role in determining the thermal performance of the heat pipe. In this regard, various types of wick structures have been developed, such as mesh screen wick, grooved wick, and sintered particle wick. The thermal performance of heat pipe is generally determined by capillary limit, which can be readily predicted based on simple analytic method represented by Equation (6). Boiling limit is also important in high operation temperature. However, a definite model for the boiling limit is still not available. The heat pipe design starts with working fluid selection, followed by wick type and container material selections, determining diameter and thickness, wick design, and heat sink–source interface design. The application of heat pipe to electronics cooling can be classified by the configuration: heat pipe–embedded spreader, block-to-block, block-to-fin, and fin-to-fin applications.

Author details

Chan Byon

Address all correspondence to: cbyon@ynu.ac.kr

School of Mechanical Engineering, Yeungnam University, Gyeongsan, South Korea

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