

Numerical Analysis of an Advanced Vapor Chamber-Heat Sink Design Embedded with PCM

Amged Al Ezzi^{†‡}, Mohammed Jawad Moahmed^{†*} & Mahmoud Mustafa Mahdi[†]

[†]Electromechanical Eng. Department, University of Technology, Baghdad, Iraq

[‡]Department of Mechanical & Aerospace Engineering, University of Missouri, Columbia, USA.

*E-mail: 50024@uotechnology.edu.iq

ABSTRACT: In recent years, with the increasing heat dissipation of electrical appliances, efficient cooling systems are required to solve the elevated temperature problems encountered in certain interior areas of electrical devices. This paper presents a new advanced proposed vapor chamber design combined a phase change material (PCM) with an extended surface area (fins). The new approach is utilizing water as working fluid, pool boiling employed as evaporation progress, and the copper foam and PCM-HS29P as a thermal storage. In order to investigate the proposed design performance, a numerical model and simulation analysis were conducted to remove 225W of heat generated from an electronic chip sized 2.5x2.5 cm² and maintain the chip temperature below 70oC for a normal operation. It was determined that when the new design outreaches 225W, the temperature of the chip is just 335.5K. Furthermore, the safety of the chip is increased with PCM used which can transfer 22% more heat input to 275W. Due to the absence of the wick structure fabrication in this approach, it is foreseeable the design cost is reduced as long as the pool boiling is employed as evaporation progress. Results obtained from this study will pave the way that will allow designing an effective new vapor chamber approach.

KEYWORDS: Phase Change Material; Heat Sink; Cuckoo Search Technique, Heat Transfer, Numerical Analysis.

INTRODUCTION

The demand for reliable high-performance electronic components is accelerating as long as the technology leading contest between electronic companies is on fire. Therefore, meeting the requirements to have high speed computing and good power density chips is in an updating [1]. However, heat generation which is a function of the electronic chip performance is considered as a main barrier that can damage the evolution of the electronic equipment technology. Both the performance reliability and life expectancy of electronic equipment are inversely related to the component temperature. One of the most common electronic chip cooling techniques is heat sinks attached to the electronic components to extend surface area and transfer the heat generation from to where forced convective air cooling is taking place; this technique knowns as low cost and simplicity. This traditional cooling technique reaches its limit as the electronic chips heat generation is continuing to increase. From here on, maintaining the working component temperature within design limit range is crucial [2].

For decades, researchers have been working to find thermal solutions that can control the design performance needs. Liquid cooling technologies including single-phase liquid cooling in micro channels, immersion flow boiling, spray cooling, jet impingement cooling, and phase change (heat pipes and vapor chamber) are topped the list of the heat sink thermal managements and heat flux removal techniques [3-5]. Amid of these techniques, a phase change heat transfer method provides the ability to disseminate high heat transfer rate from a small source and transfer to larger area through vaporization and condensation of working fluid inside [6-9] The working principle is that the working fluids are absorbing heat at the connection area and then released it when touch a cooler wall and return into evaporator through a wick structure. Both heat pipe and vapor chamber have same mechanism, but in context it is more to use heat pipe as the working function is to transfer heat from one side to another and more likely to use vapor chamber as the heat is spreading from a concentrated source over a large area. However, the characterization of the heat pipe is long with small side dimensions while vapor

chamber is short with big side dimensions.

Recently, the vapor chamber heat sink technique has the most attention due to its capability to be improved by combining with other ingenious thermal management ideas. In details, the mechanism of the vapor chamber is as follows: the heat generation source is being attached to the vapor chamber small side the evaporator where heat is absorbed, and phase change is taking place as the evaporator working fluid is vaporized. Due to pressure difference, the vapor is spreading toward the cool large side of the vapor chamber, the condenser at which phase change occurs and vapor is condensed. However, this process would not be done without a wick structure coated the entire internal surface of the vapor chamber. Because of the phase change mechanism, the vapor chamber technique presents a potential balance between high heat removal capability and preserving an isothermal heat sink base.

After the first vapor chamber approach in 2006, Koito et al. [3], several studies have investigated and analyzed the vapor chambers design and application. Hsieh et al. [4] experimentally and numerically investigated the spreading thermal resistances and the thermal performance of vapor chamber. Go [10] introduced an experimental study that showed the effects of heating power and tilt angle on the thermal resistance of a vapor chamber. Results illustrated that the vapor chamber works effectively under high heat-flux density conditions. Boukhanouf et al. [11] investigated the thermal performance of a vapor chamber using an infrared thermal imaging camera. Results showed that the thermal spreading resistance of the vapor chamber was lower than that of the solid copper and the defective vapor chamber. Chen et al. [5] introduced a numerical investigation of a whole set of thermal modules, using a plate-fin heat sink embedded with a vapor chamber. Li et al. [12] analyzed effect of a shield on the thermal and hydraulic specifications of plate-fin vapor chamber heat sinks under cross flow cooling. Li et al. [13] studied the effects of shield on thermal-fluid performance of vapor chamber heat sink. The experimental results showed that the highest surface temperature of the vapor chamber heat sink is effectively decreased by adding the shield. Songkran Wiriyasart et al. [14] investigated the thermal performance of vapor chamber by coating mini-channel heat sink with porous sintering media. Results showed 20 % higher thermal performance for the vapor chamber with porous sintering wick sheet inside than the one without. Songkran Wiriyasart et al. [15] presented a study to determine the maximum filling ratio to reach minimum thermal resistance of the vapor chambers with various configuration structures. From the experiments, the maximum filling ratio to approach the lowest thermal resistance depends on the operating power, configuration, material, and internal structure of the vapor chamber. Yong Tang et al. [16] integrated a developed vapor chamber to high-power light-emitting diodes as a thermal management technique. Features of the new vapor chamber is that it designed with parallel and orthogonal microgrooves. The new design was compared with the conventional one, and the results showed that the conjunction temperature of the developed is always lower which reflects better performance for the diodes. Jason Velardo et al. [17] experimentally investigated a vapor chamber heat spreader with hybrid wick structure. The vapor chamber was improved by a screen mesh wicks on the evaporator and condenser. The prototype experienced difficulties in startup at low heat inputs. However, the prototype results showed an ability to spread heat more effectively than the conventional copper spreader. Mengyan Wang et al. [18] presented two types of groove wick structure radial groove and cross groove. The thermal performance was evaluated, and results demonstrated that the thermal spreading resistance showed an important role in the total thermal resistance of the vapor chamber heat spreader. Vanessa Michels et al. [19] investigated a new vapor chamber prototype designed with hollow fins in which the generated vapor at the base streams up to the top of the fins. Results showed 20% less overall thermal resistance than traditional pin fin heat sink.

In light of the literature review, the vapor chambers heat sink technique shows an excellent heat dissipation capability for the electronic components. Factors such as material, fabrication, and design ideas show a significant effect on the heat removal performance. In spite of the advantage of the vapor chamber technology, the most disadvantages of these techniques are the fabrication cost and work range limitation which the designers must be carefully considered all. However, to overcome this problem, an innovation fabrication and combinations have been utilized to enhance heat dissipation capability. In this work, a new advanced proposed vapor chamber design combined a phase change material (PCM) with fins is presented. A numerical model and simulation analysis were conducted. The numerical results obtained from this study are expected to lead to guidelines that will allow designing an effective new vapor chamber approach.

VAPOR CHAMBER WITH PCM

The conception being proposed here is to utilize the high capability of the PCMs as an additional heat storage which absorbs and releases heat when it melts and solidifies, respectively. In this design, the vapor chamber sides will be surrounded by the PCM which in turns will be connected to an extended surface area, fins. Figure 1 shows a schematic of the design concept. The vaporized working fluid at the base of the vapor chamber will exchange heat with the PCM and condense back to complete the circle.

Based on the thermodynamic principles, the highest heat dissipation occurs as the phase change is taking place. This amount of removal heat is equivalent to the latent heat of diffusion and solidification of the materials. Paraffin wax, metal hydrates, and fatty acids are examples of PCMs that can be used for latent thermal energy storage. However, during a phase change the temperature of the material remains constant means that a phase change process involves a large amount of heat transfer at a constant temperature; both are attractive features for heating, cooling, and temperature stabilization applications. According to that, using the PCMs in a sophistication amount will improve heat output absorption, keep the temperature constant, and enhance the safety of the vapor chamber design.

The main objective of this work is to demonstrate that the new approach has the capability to effectively and safely maintain the temperature of an electronic chip under the design limit. Therefore, the proposed design will be tested under a case of study in which meets the needs of the developers of the electronic chips. Next coming sections will illustrate the design sections and analysis.

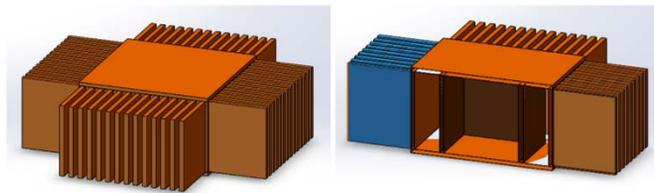


Figure 1: Schematic of the vapor chamber with PCM and fins

Design Characterizations

Due to its desirable thermal conductivity property, copper is the most likely choice to make the vapor chamber body and fins. However, this new vapor chamber has been approached without utilizing a wick structure, and that may effectively reduce the manufacturing cost of the whole system. Distilled water has been selected as a working fluid, and pool boiling evaporation progress is applied. Performance of the proposed design has been tested to remove 225W of heat generated from an electronic chip sized 1x1 in², and maintain the chip temperature below 70oC for a normal operation. The geometrical parameters of the design are listed in Table 1 and 2. The copper thickness of the vapor chamber frame and fins is 0.003m and 0.002m, respectively.

Table 1. Dimensions of the vapor chamber

	Vapor Chamber	PCM Area	Fins
Length (m)	0.06	0.06	0.06
Width (m)	0.06	0.02	0.05
Height (m)	0.06	0.06	-
Gap (m)	-	-	0.002
Number of Fins	-	-	14x4

For the thermal energy storage PCM, A promising composite material made of copper foam and PCM-HS29P is chosen. The ratio of Copper and PCM material is 50%. Copper foam is a product of Duocel® Copper Foam, which is already widely used in industry. PCM-HS29P is a product provided by RGEES LLC. It's uniquely engineered mixture of hydrated salts with high capacity to store thermal energy as latent heat. It shows good thermal stability, thermal conductivity, and retain its latent heat without any change in physical

or chemical properties for over thousands of cycles [20]. Table 2. Shows the Properties of PCM-HS29P.

Table 2. Properties of the PCM-HS29P [20]

Property	Value
Melting Temp. (°C)	29 T
Freezing Temp (°C)	29 T
Liquid Density (kg/m ³)	1550
Solid Density (kg/m ³)	1840
Latent Heat (kJ/kg)	190
Specific Heat-Liquid (kcal/kg·K)	0.54
Thermal Conductivity (W/m·K) Liquid	0.54
Thermal Conductivity (W/m·K) Solid	1.09
Base Material	CaCl ₂
Congruent Melting	Yes
Sub Cooling	No
Flammability No -	No
Thermal Stability (cycles)	5000 Internal
Max. Operating Temp (°C)	80

The design will utilize a cooling fan to enhance heat dissipation. Heat transfer effectiveness, size, and noise output are the main concerns when choosing a fan. The heat sink demands 1.8 CFM of air flow from the fan. The fan must fit within the heat sink design; therefore, no frame dimension may exceed 25 mm. For comfortable operation, the fan should not exceed 30 dB. The ADDA AD2506fan from Shenzhen the Haixing and Electronics Company will be implemented in the design to meet all the requirements. Properties and specifications of the fan may be found in Table 3.

When using forced convection, or powered air flow, cooling is further enhanced by many heat sink characteristics. Ducting keeps the air flow in the heat sink and limits by-pass air. Fins should be oriented so as not to prevent air flow. The heat sink should be optimized for airflow and pressure drop so as not to put back pressure on the fan which could damage it. Lastly, wider heat sinks are usually more favorable than taller or longer designs.

Table 3. ADDA 25x25x10 mm DC Cooling Fan Specs. [21]

Property	Value
Frame Size (mm)	25x25x10
Rated Voltage	5V, 12 V
Rated Speed	6,000~10,000
Max Air Flow	1.6~2.1 CFM
Max Noise	18.5~23 dB
Life Expectancy (Ball Bearing)	70,000 hrs
Life Expectancy (Hypro Bearing)	>5,000 hrs
Operation Temperature	-10~75 °C
Storage Temperature	-40~75 °C

NUMERICAL ANALYSIS RESULTS

As prementioned in the design parameters, the input power of the chip, chip area, and vapor chamber area are 225W, 0.025x0.025 m², and 0.06x0.06 m², respectively. The heat flux can be found from

$$q_{in} = \frac{Q_{in}}{A} \quad (1)$$

The following assumptions are made for the numerical analysis model:
 $x = 0.06 \text{ m}$. $y = 0.02 \text{ m}$. $z = 0.06 \text{ m}$. $t = 0.003 \text{ m}$. $T_{air} = 293K = 20 \text{ }^\circ\text{C}$

Basic information of the base part and fins can also be found in Table 1. The heat transfer process might be characterized by a group of thermal resistances, shown in Figure 2, signifying heat charging and discharging modes as follows:

- R1 - liquid-vapor phase change process resistance
- R2 - vapor-condensate interfacial resistance
- R3- condensate film resistance
- R4 - column wall resistance (lateral direction)
- R5 - PCM conduction resistance with foam
- R6 - column wall resistance (lateral direction)
- R7 - resistance of fin outside of PCM

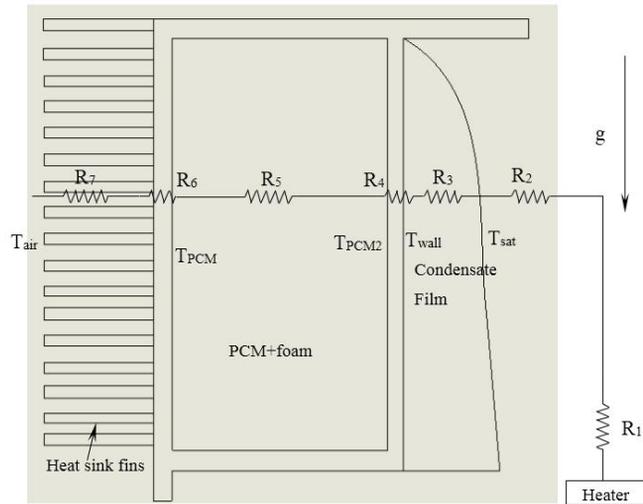


Figure 2. Thermal resistances network of the vapor chamber

The number of fins is obtained as

$$N = \frac{W_b}{s+t_f} \quad (2)$$

The total volume of heat sink is

$$V_b = t_b x w_b x l_b \quad (3)$$

In order to enhance the heat transfer in fins, forced convection is added with certain fans or pumps. The ADDA DC cooling fan is recommended with a volume flow rate 1.8CFM, Table. 3, which equals to $8.4942 \times 10^{-4} \text{ m}^3/\text{s}$. Thus, the air flow velocity is $v_a = 8.4942 \times 10^{-4} \text{ m}^3/\text{s} \div (0.01 \times 0.01 \text{ m}^2) = 8.4942 \text{ m/s}$.

The correlated parameters are listed in Table 4. Except for the PCM foam part, the whole model is made of copper with thermal conductivity $k_{\text{wall}} = 400 \text{ W} / (\text{m.K})$.

Table 4. Parameters of air

Property	Value
Thermal conductivity of air	$K_a = 0.0263 \text{ W}/(\text{m.k})$
Density of air (Kg/m^3)	$\rho = 1.1614 \text{ kg}/\text{m}^3$
Dynamic viscosity of air	$\mu = 184.6 \times 10^{-7} \text{ N.s}/\text{m}^2$
Prandtl number (Pr)	$\text{Pr} = 0.707$

Convective heat transfer coefficient

Fins are cut into 10 sections in length to achieve a higher convection coefficient. So, the effective length of each fin is

$$l_{ba} = l_b/10 \quad (4)$$

Reynolds number of the air flowing through gap is

$$Re = \frac{\rho v_a l_{ba}}{\mu} = 3206.4 > 2300 \quad (5)$$

The corresponding Nusselt number is

$$Nu = 0.664 Re^{1/2} Pr^{1/3} \quad (6)$$

The average convection coefficient acting on the fins is

$$h_f = \frac{k_a \cdot l_{ba}}{Nu} \quad (7)$$

The thermal resistance of the fin is obtained in the following process: the adjusted length of fins is

$$l_{fn} = l_f + \left(\frac{t_f}{2}\right) \quad (8)$$

surface area of the base is

$$A_{sb} = (l_b * w_b) - (N * (t_f * l_b)) \quad (9)$$

perimeter of fins is

$$P = t_b + 2 * l_{fn} \quad (10)$$

cross sectional area of fins is

$$A_{cf} = t_f * l_b \quad (11)$$

surface area of the fin adjusted for adiabatic tip is

$$A_f = 2 * l_b * l_{fn} \quad (12)$$

total surface area of a fin surface array is

$$A_t = N * A_f + A_{sb} \quad (13)$$

fin efficiency can be found

$$N_f = \frac{\tanh(m_l)}{m_l} \quad (14)$$

Where

$$m_l = m * l_{fn} \quad (15)$$

and

$$m = \sqrt{\frac{2P}{k_{wall} * A_{cf}}} \quad (16)$$

the overall efficiency is

$$N_o = 1 - \frac{N * A_f}{A_t} * (1 - N_f) \quad (17)$$

Finally, the thermal resistance of the overall fin array is

$$R_7 = \frac{1}{N_o * h_f * A_t} \quad (18)$$

Resistance of the base of the heat sink is

$$R_6 = \frac{t_b}{k_{wall} * (w_b * l_b)} \quad (19)$$

Temperature of the PCM is then achieved by

$$T_{PCM} = T_{air} + \frac{Q_{in}}{2} * (R_6 + R_7) \quad (20)$$

Then, the thermal resistance of the PCM foam is

$$R_5 = \frac{y}{k_{PCM} * x * z} \quad (21)$$

Where, the thermal conductivity of the PCM foam is $k_{PCM} = 200.5 W/(m.K)$

Then, wall resistance is

$$R_4 = \frac{t_{wall}}{k_{wall} * x * z} \quad (22)$$

The related temperature can be found:

$$T_{PCM2} = T_{PCM} + q_i * R_5 * x * z \quad (23)$$

$$T_{wall} = T_{PCM} + q_i * (R_4 + R_5) * x * z \quad (24)$$

In order to get thermal resistance of the liquid-vapor part in the chamber, assumption of T_v is needed

The first assumption is $T_v = 305K$; therefore, the relative parameters of water is $\rho_l = 993.7 kg/m^3$, $\rho_v = 0.0498 kg/m^3$, $k_l = 622.525 W/(m.K)$, then the saturation temperature can be obtained

$$T_{sat} = T_{wall} + R_3 * q_i * x * z \quad (25)$$

Then, the R3 and R2 are calculated as follow:

$$R_3 = \left(\frac{0.79 \times 4 \times z_3 \times g \times \rho_{wl} (\rho_{wl} - \rho_{wv}) k_w^3 h_{lv}}{\mu_l (T_{sat} - T_{wall})} \right) \quad (26)$$

$$R_2 = \left[\left(\frac{2 \rho_{x,z}}{2 - \rho} \right) \left(\frac{h_{lv}}{T_{sat} v_{lv}} \right) \left(\frac{M}{2 \pi R_u T_{sat}} \right) 0.5 \left(1 - \frac{P_{sat} v_{lv}}{2 h_{lv}} \right) \right]^{-1} \quad (27)$$

Where, $P_{sat} = 5.6$, $M = 1.8 \times 10^{-4}$, $\bar{R} = 8.31$ J/mol.K

With the assumption, the real T_v is found, if it's not different from the assumption, a new number is guessed for another iteration.

After finding the accurate number of T_v , the temperature of the chip can be found

$$T_{boil} = T_v + Q_{in} * R_2 \quad (28)$$

$$T_c = T_{boil} + q_{in}^{0.327} / C_1 \quad (29)$$

Where, T_{boil} is the temperature due to pool boiling and the coefficient to get chip temperature is $C_1 = 2.5$. Eq. 29 is derived from Stephan and Abdelsalam correlation of water $q'' = (C_1 (T_w - T_{sat(pl)}))^{1/0.327}$ in the nucleate region.

The boiling limitation is then examined by calculating the super heat of pool boiling

$$dT = T_h - T_{boil} \quad (30)$$

Numerical calculation results

Results of the numerical model are listed in Table 5 below:

Table 5. Calculation results

Property	Value
Heat flux	62500W / m ²
The number of fins	15
Total volume of heat sink	1.08×10 ⁻⁵ m ³
Effective length of each fin	0.006m
Nusselt number	33.4955
Average convection coefficient	146.8219W / (m ² · °K)
Adjusted length of fins	0.051m
Surface area of base	0.0018m ²
Perimeter of fins	0.105m
Cross sectional area of fins	1.2×10 ⁻⁴ m ²
Surface area of the fin adjusted for adiabatic tip	0.0061m ²
Total surface area of fin array	0.0936m ²
Fin efficiency	0.9962
Overall fin efficiency	0.9963
Temperature of PCM	301.45K
Temperature of PCM inner side wall	304.57K
Temperature of wall	304.80K
Saturation temperature of water (phase change medium)	305.97K
Temperature of pool boiling	325.80K
Temperature of chip	340.60K

Temperature difference (Superheat) 14.80K

The thermal resistances result of the numerical model are illustrated in Table 6. However, R1 has been neglected due to its small value which approximately considered zero.

Table 6. Numerical model thermal resistance results

R _{th}	Value
Resistance of the overall fin array	R ₇ = 0.0730 W/(m ² ·K)
Resistance of the base of the heat sink	R ₆ = 0.0021 W/(m ² ·K)
Thermal resistance of PCM foam	R ₅ = 0.0277 W/(m ² ·K)
Resistance of wall	R ₄ = 0.0021 W/(m ² ·K)
Condensate film resistance	R ₃ = 0.0104 W/(m ² ·K)
Vapor-condensate interfacial resistance	R ₂ = 0.0924 W/(m ² ·K)

RESULTS ANALYSIS

Conducting the numerical model showed promising results that focus on maintaining the chip temperature within normal working range and providing a safety factor for the overheating cases. Nevertheless, the results demonstrate the PCM role in chip protection by absorbing the unexpected heat. Figure 3 below, shows that when the vapor chamber heat sink outreaches 225W, the temperature of the chip is just 335.5K; it bellows the design required temperature 70oC (343K). Thus, the design is satisfied needs. However, as the heat output is continuously increasing, the chip temperature will keep evolving in a linear pattern. Therefore, the heat sink keeps on working to remove heat until the PCM material began melting. At this time, the output of the chip is already 275W and the chip is still below 70°C, which is 68°C. In conclusion, the proposal design meets the design requirement with a safety factor [22-25].

$$\eta = \frac{275-225}{225} = 22\%$$

(31)

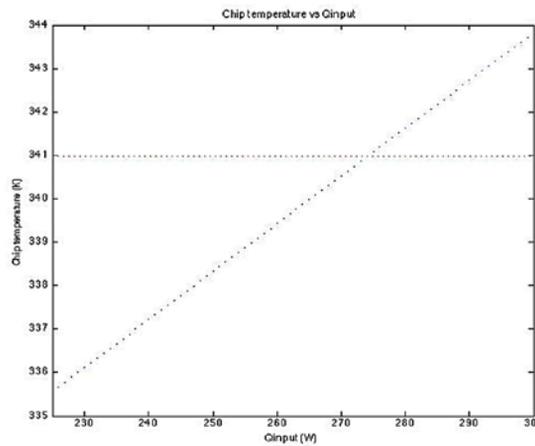


Figure 3. Chip temperature versus heat input

With this safety factor, the heat sink would keep the chip safety with a 50W more output. Figure 4 indicates that when the working out is 225W, the temperature of PCM material is just 300.7K, the melting temperature of PCM is 302K, the PCM would begin to melt when the output of the chip reaches 275W.

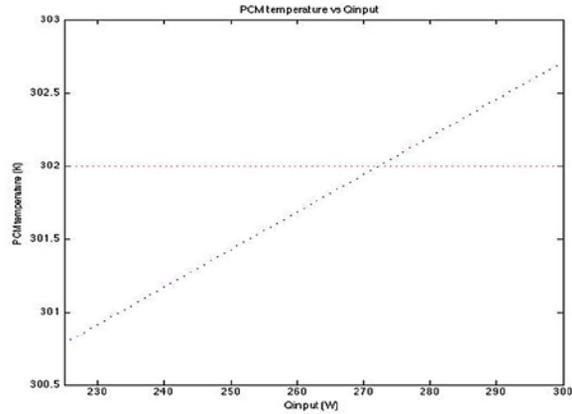


Figure 4. PCM temperature versus heat input

Figure 5 describes the time progress of the melting process of the PCM from solid to liquid phases. It takes 33 minutes to totally melting and become 100% liquid. Within the melting period, the PCM would keep changing phase and absorbing heat; the heat storage ability is 100kJ. Therefore, the temperature of the whole system will be invariant. For these 33 minutes, the chip would obviously be safe. Using PCM material is a key point of the design approach by adding extra protection to the chip from overheating.

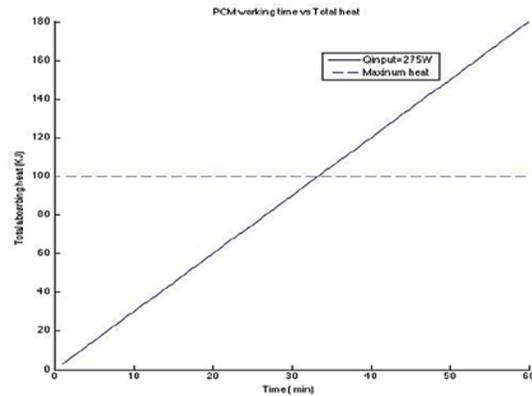


Figure 5. PCM working time of total melting

Figure 6 shows the pooling boiling limitation of this vapor chamber-heat sink. Since there is no an equation developed in other pressure, the equation used in the numerical analysis is pooling boiling equation in standard atmospheric pressure. Obviously, this figure illustrates that the superheat would increase as output is increased, but the difference is only 0.35K, which means superheat is mainly no change.

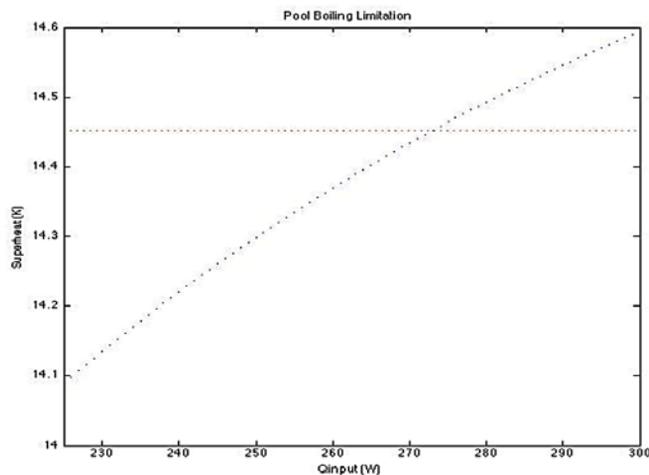


Figure 6. Pool boiling limitation versus heat input

For the numerical analysis process, thermophysical properties for saturated liquid and vapor are updated whenever T_v are changed in the process of iterative optimization because of varying other variables. Also, when the assumptions don't meet requirements of certain limitations, new numbers should be guessed and then iterate again. Thus, optimization is significantly important to get an accurate answer. An optimization flow-chart is shown in Figure 7.

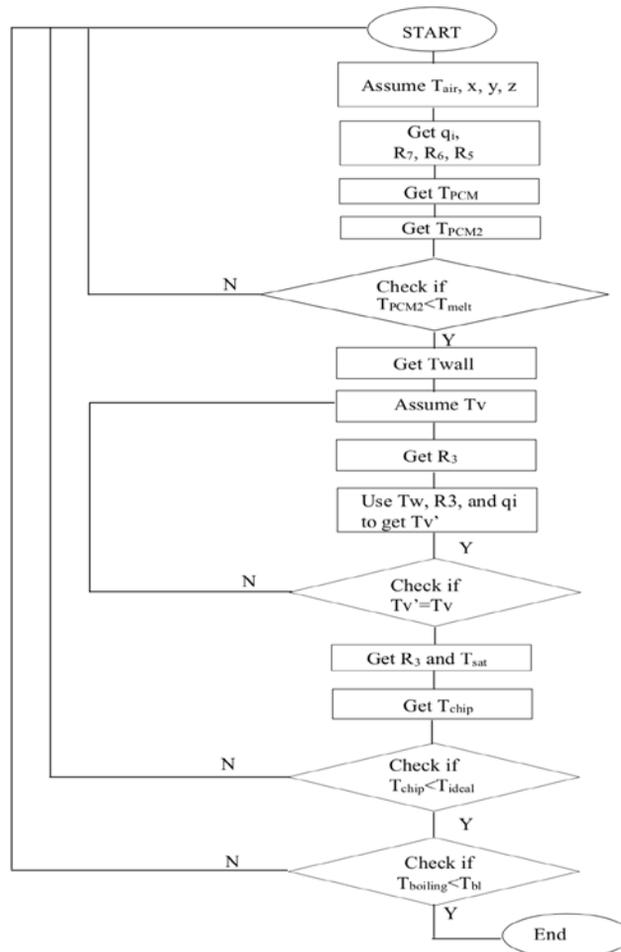


Figure 7. Optimization flow-chart of the vapor chamber-heat sink model

Numerical simulation

Heat transfer simulation was conducted by using ANSYS Steady State Thermal module. Comparing to analytical model, numerical model extends the numerical model into 3D. The pool boiling area is surrounded by PCM, and the whole structure is covered in the copper. The size is same as analytical model. Figure 8 is a section view of the whole structure. However, top cover in Figure 9 is suppressed to show the inner structure. Heat is added at the center of the bottom copper cover, with a total heat flow of 225W. Convection heat transfer is applied at four side walls to extract heat from the heat sink.

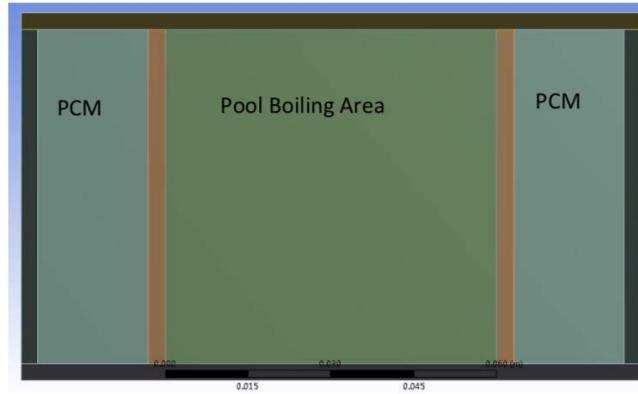


Figure 8. Section view for the vapor chamber-heat sink

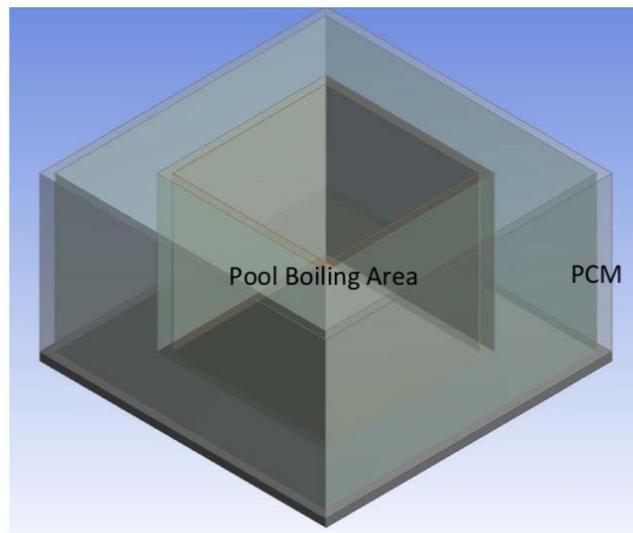


Figure 9. 3D view of the vapor chamber-heat sink

Several simplifications are made in this model:

1) Fin on the outer wall is ignored by increasing heat transfer coefficient using equation, so that temperature drop across the outer copper wall is the same as the one with fin on it. As in the analytical model.

$$h_{eff} = h_0 \frac{A_{fin}}{A_{eff}} \quad (32)$$

Where, h_{eff} is effective heat transfer coefficient, h_0 is heat transfer coefficient with fin on the surface, A_{fin} is surface area with fin, A_{eff} is surface area without fin.

2) Pool boiling in the center of the structure is simulated by heat conduction, thermal conductivity is tuned to match the temperature difference from pool boiling and film condensation.

3) All interface resistance is ignored in this model, the value of contact thermal conductance is calculate using the smaller thermal conductivity of the two material in contact.

Mesh dependency

Three different meshes are used to test the grid dependency. Coarse mesh with 92182 nodes, 18619 elements; medium mesh with 261998 nodes, 55754elements; fine mesh with 515635 nodes, 113854 elements. The temperature result is listed in Table 7.

Table 7. Results of different meshes used

Mesh Type	Maximum Temperature	Minimum Temperature
Coarse	58.673	28.919
Medium	58.724	28.919

Listed results demonstrate that comparing to fine mesh, the coarse mesh tends to underestimate the temperature, and the medium mesh tends to overestimate the temperature, but the difference between meshes is negligible, because the whole problem does not contain any transient or phase change problem. Analysis afterwards will base on the result from fine mesh.

Temperature Result

Figure 10 shows the temperature distribution inside the heat sink, this is the section view from the center of the heat sink, and we can see the heat is uniformly spread in the heat sink. Such a distribution is due to conduction assumption. The real temperature difference will concentrate at the bottom surface and side wall, which is the pool boiling area and film condensation area, Figure 11. With that kind of temperature distribution, it is expected that the heat transfer will be better compared to the numerical model; since pool boiling and film condensation will transfer heat more uniform from the chip to PCM.

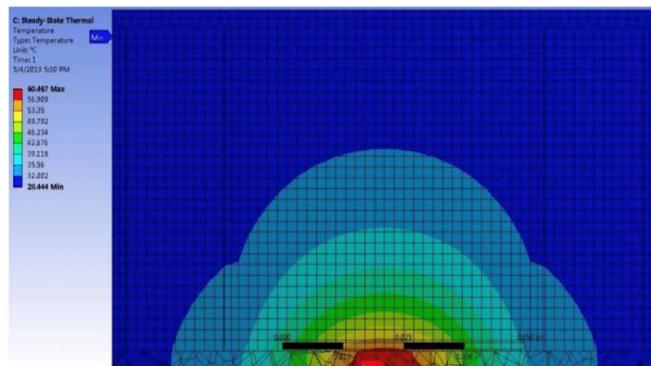


Figure 10. Section view of the temperature distribution inside the vapor chamber

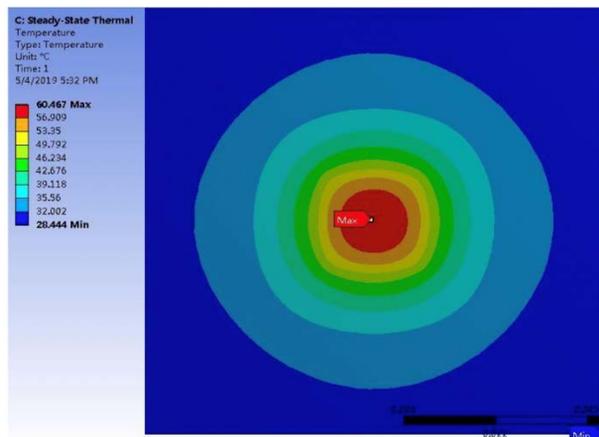


Figure 11. Bottom view of the vapor chamber temperature distribution

Fig. 11 Bottom view of the vapor chamber temperature distribution

Figure 11 is the bottom temperature distribution. The figure is illustrated that the high temperature region is concentrate at the center of the bottom, where the power is added. Pool boiling in the real situation will spread the heat more uniformly to the side.

One very important temperature distribution is the temperature distribution in the effective phase change area. The effective thermal conductivity is determined by matching the temperature drop inside phase change section with the real temperature drop across the pool boiling area and film condensation area. Using this setup, the temperature difference in the phase change is 29 degree, which is similar to the numerical result; the temperature drops across the pool boiling area is 16 degree, and across film condensation is 9 degree, see Figure 12.

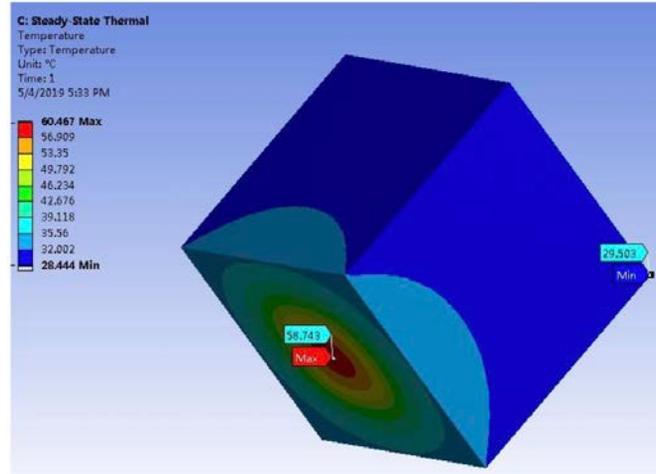


Figure 12. Temperature distribution within the PCM

Figure 13 shows the temperature distribution in the PCM. Two labels on the figure show the highest and lowest temperature inside PCM. From the figure, the highest temperature is slightly lower than the melting temperature. However, the graphite foam greatly helped PCM to achieve a uniform temperature difference.

The numerical simulation model showed that the heat sink can keep the highest temperature on the chip under 70-degree Celsius and keep the highest temperature in PCM lower than the melting point. This simple numerical simulation is capable of extract 225W power from the chip.

The numerical now is a simplified model, no transient solution is provided in this model, a more accurate model can be created considering pool boiling and film wise condensation. However, due to the limited capability of ANSYS software on calculate melting and solidification, an effective liquid-vapor phase change may be practical to predict the heat transfer performance. User defined function (UDF) may be needed in order to describe the properties of the effective liquid-vapor phase change material.

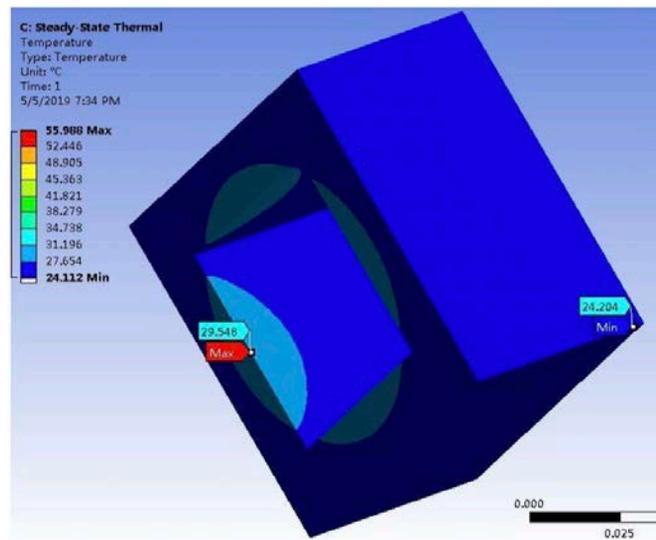


Figure 13. Temperature distribution within the PCM

CONCLUSION

A new advanced proposed vapor chamber design combined a phase change material (PCM) with fins is presented. A numerical model and simulation analysis were conducted. The new approach has shown that it is possible to have a heat sink with attractive features like high heat storage ability, fast charging ability,

environment safe operation, high heat flux capability and high energy storage density including added features of a vapor chamber like self-acting and quiet operation. The promising aspects that make this concept feasible include using carbon foam in the thermal energy system columns (which help the thermal conductivity of heat to increase from 10.5 W/(m·K) to 205.5 W/(m·K), which helps in fast charging/heat absorption and a low heat flux at each column thus reducing the temperature gradients. Using of copper foam inside the PCM material of the chamber to ensure rapid transport of heat loads. With a safety factor 22% and heat storage ability 100kJ would give double assurance of the chip safety. Due to the absence of the wick structure fabrication in this design, the cost is reduced as long as the pool boiling is employed as evaporation progress.

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