Faculty of Mechanical Engineering

GEOMETRIC OPTIMIZATION OF LED HEAT SINK FOR COOLING OF LED LIGHTING

Hor Woei Tsong

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GEOMETRIC OPTIMIZATION OF LED HEAT SINK

FOR COOLING OF LED LIGHTING

HOR WOEI TSONG

A thesis submitted

in fulfillment of the requirements for the Bachelor of Mechanical Engineering with Honours

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2018

DECLARATION

I declare that this project report entitled "Geometric optimization of the led heat sink for cooling of led lighting" is the result of my own work except cited in the references.

Signature	:	....
Name	:	HOR WOEI TSONG
Date	:	....

SUPERVISOR'S APPROVAL

I hereby declare that I have read this project report and in my opinion, this report is sufficient in terms of scope and quality for the award of the Bachelor of Mechanical Engineering with Honours.

Signature : ……………………………………………………………………

Name : DR. CHENG SEE YUAN

Date : ……………………………………………………………………

ABSTRACT

Light Emitting Diode(LED) lighting is one of the well-known energy efficient and rapidly developing technologies. LED lighting has a longer lifespan, higher durability and better light quality if have a good cooling performance of heat sink. The purpose of this research is to optimize the geometry of heat sink without increasing production cost to achieve high performance of heat sink without using excessive materials. The experimental work of the study will be expensive. Therefore, software simulation Ansys version 16 was used to simulate the case study. In order to validate the simulation were correct, comparison of the experimental and numerical result was done. Once the difference between them was not huge, case study proceeded. After several case studies were done, it found out that reduce in fin thickness of heat sink and increased in height of fin of heat sink had the highest cooling performance of heat sink compared to the others. This is because reduce in fin thickness resulted the thermal boundary layer did not fully develop at an early stage. While increase in fin height resulted in a huge increase in surface area, therefore, heat transfer surface area greatly increased. This study will be used for geometry optimization of the heat sink with another shape.

ABSTRAK

Pencahayaan Diod Pemancar Cahaya (LED) merupakan salah satu teknologi yang cekap tenaga dan teknologi yang berkembang pesat. Pencahayaan LED mempunyai jangka hayat yang lebih panjang, ketahanan yang lebih tinggi dan kualiti cahaya yang lebih baik jika adanya penyejuk haba yang prestasi baik. Tujuan penyelidikan ini adalah untuk mengoptimumkan geometri sinki haba tanpa meningkatkan kos pengeluaran supaya mencapai prestasi sinki haba yang baik tanpa menggunakan bahan berlebihan. Oleh sebab kerja eksperimen kajian mahal. Oleh itu, simulasi perisian Ansys versi 16 digunakan untuk mensimulasikan kes kajian. Supaya pengesahan simulasi adalah betul, perbandingan keputusan eksperimen dan simulasi dilakukan. Sebaik sahaja perbezaan antara mereka tidak besar, kajian kes terus dijalankan. Selepas beberapa kajian kes dilakukan, ia mendapati bahawa mengurangkan ketebalan sirip sink haba dan peningkatan ketinggian sirip haba mencapai prestasi penyejukan yang paling tinggi daripada sinki haba yang lain. Ini disebabkan dengan mengurangkan ketebalan sirip sinki haba menyebabkan lapisan sempadan termal tidak sepenuhnya berkembang pada peringkat awal. Walaupun peningkatan ketinggian sirip menyebabkan peningkatan yang besar di kawasan permukaan. Oleh itu, kawasan permukaan pemindahan haba bertambah tinggi. Kajian ini akan berguna untuk mengoptimumkan geometri sinki haba yang berbentuk lain.

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Last but not least, thanks to everyone who is involved in my final year project directly and indirectly for their help.

CONTENT

LIST OF FIGURES

LIST OF TABLES

LIST OF ABBREVATIONS

LED	Light Emitting Diode
L	Long
LM	Long Middle
LMS	Long Middle Small
CFD	Computational Fluid Dynamics

LIST OF SYMBOL

R_{th}	=	Thermal Resistance (°C/W)
h	=	Heat Transfer Coefficient
H	=	Height Of Fins (mm)
θ	=	Degree
A	=	Surface Area (mm^2)
$\dot{m}$	=	mass flow rate (kg/s)
λ	=	Thermal Conductivity (W/m.K)
E	=	Young's Modulus
α	=	Electrical Resistivity, (μΩ.Cm)
Cp	=	Specific Heat Capacity (H/g.°C)
ρ	=	Density
t	=	Thickness (mm)
$\dot{q}$	=	Heat Flux (W/m^2)
P	=	Pressure (Pa)
r	=	Radius (mm)
T	=	Temperature (°C)
M	=	Mass (kg)
L	=	Length (mm)
u	=	x-component of velocity (m/s)
v	=	y-component of velocity (m/s)
w	=	z-component of velocity (m/s)
g	=	Gravity (m/s^2)
μ	=	Dynamic viscosity (N/m^2s)
ref	=	reference
ave	=	average
o	=	outer
i	=	inner
l	=	long
m	=	middle

∞	=	ambient
f	=	fins

CHAPTER 1

INTRODUCTION

1.1 Background

Light- Emitting Diode (LED) is one of the well-known energy-efficient and rapidly- developing technologies. LED lighting has a longer lifespan, higher durability and better light quality compared to incandescent lighting and other types of lighting. Although LED lightings do have a lot of advantages, it still faces the issue of heat. Heat produced by the led lighting will cause failure toward LED lighting.

Heat is the greatest enemy of LED technology (Victor, 2015). Heat brings failure or drop of performance for LED. While the failure included light output decreased permanently due to the damage was done by the heat toward LED, the color temperature of the white LED changed and etc. This failure because of materials used in the LED unable to withstand high temperature. To prevent LED lighting from overheating, a heat sink for led lighting must be applied.

Without a good heat sink, the internal temperature of LED lighting increases causing to failure. With the increase of internal temperature of a LED, voltage and lumen output of the LED decreases. With the characteristic changed, brightness and efficiency as well as an overall lifetime of LED decreased. Internal temperature of LED high lead to faster LED deterioration. This is why it is important to make sure internal temperature of LED remains low. (Taylor Scully, 2015)

There were several challenges in heat dissipation of LED lighting using a heat sink. The heat sink is a cylinder with longitudinal fins that usually applied in LED light bulbs. With a suitable design of heat sink, it will bring out the most effective on heat dissipation of LED lighting. Which will greatly reduce the overheated of LED lighting that will cause a reduction in the performance of lighting?

Due to competitive of LED lighting in the market, LED lighting industry required to develop innovative, low-cost conductive and convection heat sink to make sure LED lighting performance last longer.

1.2 Problem Statement

With the incandescent light is fading away, LED light is taking the sports light of the current market. However, LED light bulb is facing a problem when dealing with heat dissipation due to its limitation of internal temperature that causes a drop in performance. In order to reduce the internal temperature of LED light, heatsink was applied. Although heat sink will reduce the internal temperature of LED light, heat sinks are expensive in the market. In order for the manufacturer to provide an efficient heat sink with same price. Without excessive usage of materials, optimization of the design of heat sink should be done without increasing the cost of production.

1.3 OBJECTIVE

The objectives of this project are as follows:

1. To analyze the differences in geometry of heat sink like height of fin, length of fin and thickness of fin of the heat sink effect on its conjugate heat transfer by performing simulation under natural convection.
2. To provide an improved heat sink performance after optimization of geometry without an increase in mass.

1.4 Scope of Project

The scopes of this project are:

1. Geometry optimization of LED heat sink only focuses on the radial heat sink.
2. Simulation geometry changes of the LED heat sink without an increase in mass
3. Radiative heat transfer at the heat sink is negligible.

CHAPTER 2

LITERATURE REVIEW

2.1 LED Heat Sink

Light Emitting Diode (LED) lightings replaced by conventional lighting devices which provide higher efficiency, longer life and small in size. A lifetime of LED will be reduced if did not apply heat sinks with the main purpose of cooling LEDs (Kwak et. al. 2017). Similar to (Park et. al. 2016), stated thermal dissipation structure needed to assure long life and efficiency of LED lamp.

2.1.1 Shape of Heat Sink

According (Yu et. al. 2010) stated that majority of studies related to rectangular bases of heatsinks did not produce a significant result for cooling circular LED lights. Therefore, a radial heat sink with circular base and rectangular fins were studied. There was three types model of heat sink (Yu et. al. 2011) studied which were L type model, LM type model and LMS type model. LM type model showed the most efficient in heat dissipation in the result. Figures below show L type model and LM type model of heat sink.

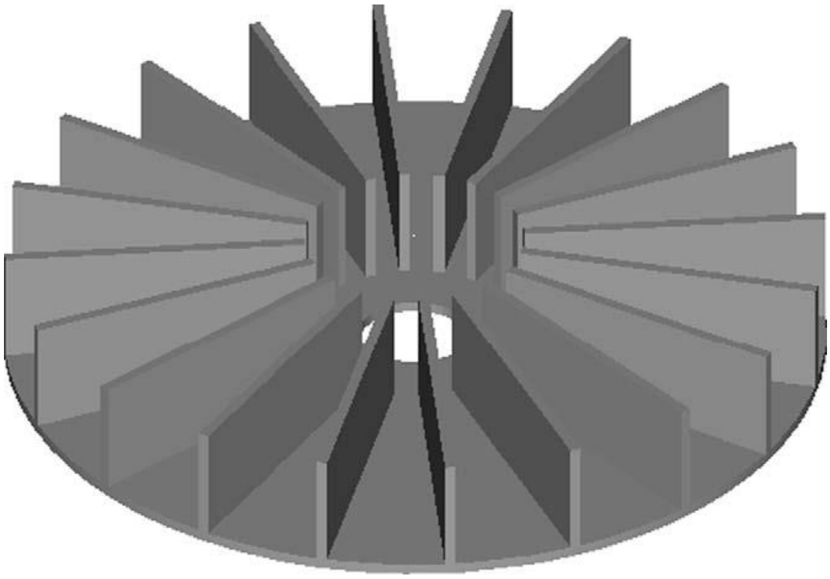


Figure 2.1 : L type model (Yu et. al, 2011).

Figure 2.2 : LM type model (Yu et. al, 2011).

2.1.2 Material of Heat Sink

According to J.Padmaja & A. Ravindra (2015), selection of material heat sink is important in the conduction of heat from the heatsink. According to G. Prashant Reddy & Navneet Guptal, (2010) studies, the high thermal conductivity of the material used for heat sink, the faster the heat dissipation to the surrounding. There was three cases study specified by them.

Table 2.1 : Condition of material selection (G. Prashant Reddy & Navneet Gupta1, 2010)

Case 1 :

Function	Heat Sink
Constraints	(1) Material must have $\rho_e > 10^{19} \mu\Omega\ cm$ (2) All dimensions specified
Objective	Maximize thermal conductivity
Free Variables	Choice of material

Based on condition, a graph in figure 2.3 was formed. The graph showed that Aluminum Nitride (AIN) or Alumina (Al_2O_3) satisfied the condition.

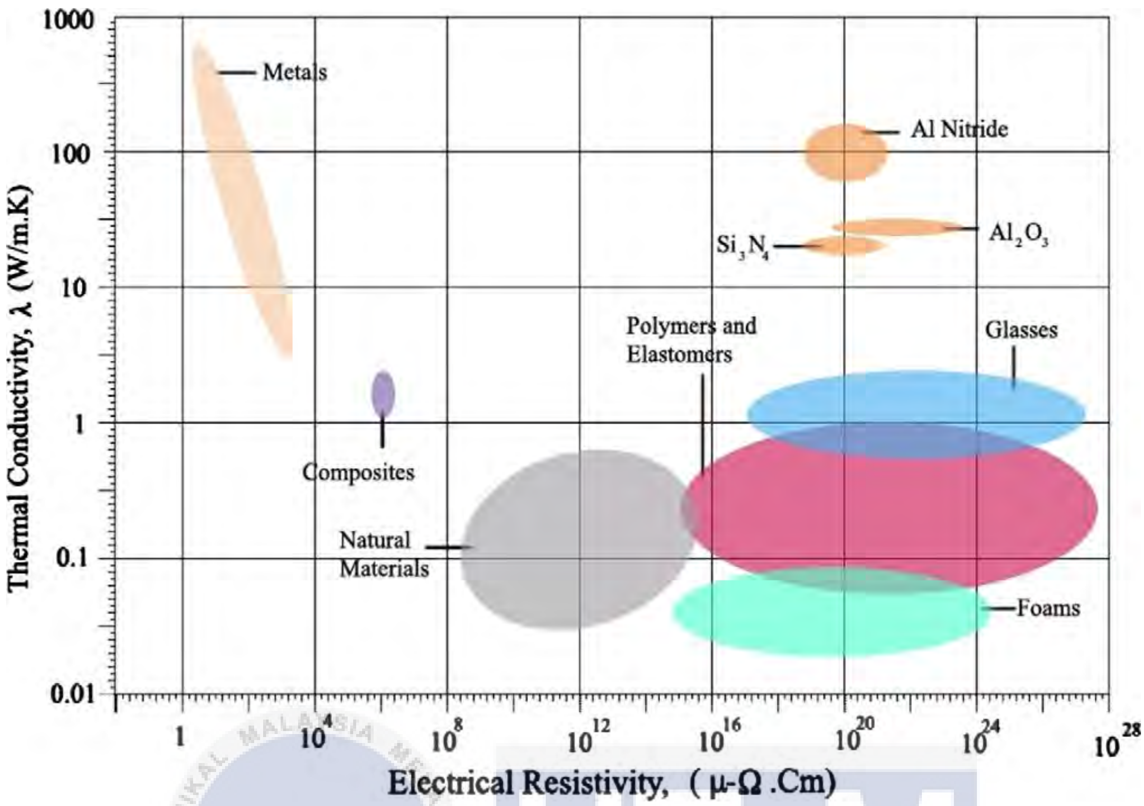


Figure 2.3 : thermal conductivity vs electrical resistivity for different class of materials (G. Prashant Reddy & Navneet Gupta1, 2010)

Table 2.2 : Second Condition of material selection (G. Prashant Reddy & Navneet Gupta1, 2010)

Case 2:

Function	Heat Sink
Constraints	(1) The temperature of material used in heat sink increase, thermal expansion decrease (2) High electrical resistivity (3) High value of Young's Modulus, thermal expansion must be significant
Objective	(1) Maximize Young's Modulus (2) Heat Transfer Coefficient increase, temperature increase

From the graph in figure 2.4, Al, AlN, Al_2O_3 fulfilled the condition for case 2. Therefore, aluminum-based alloys or metals are very optimistic materials for the heatsink.

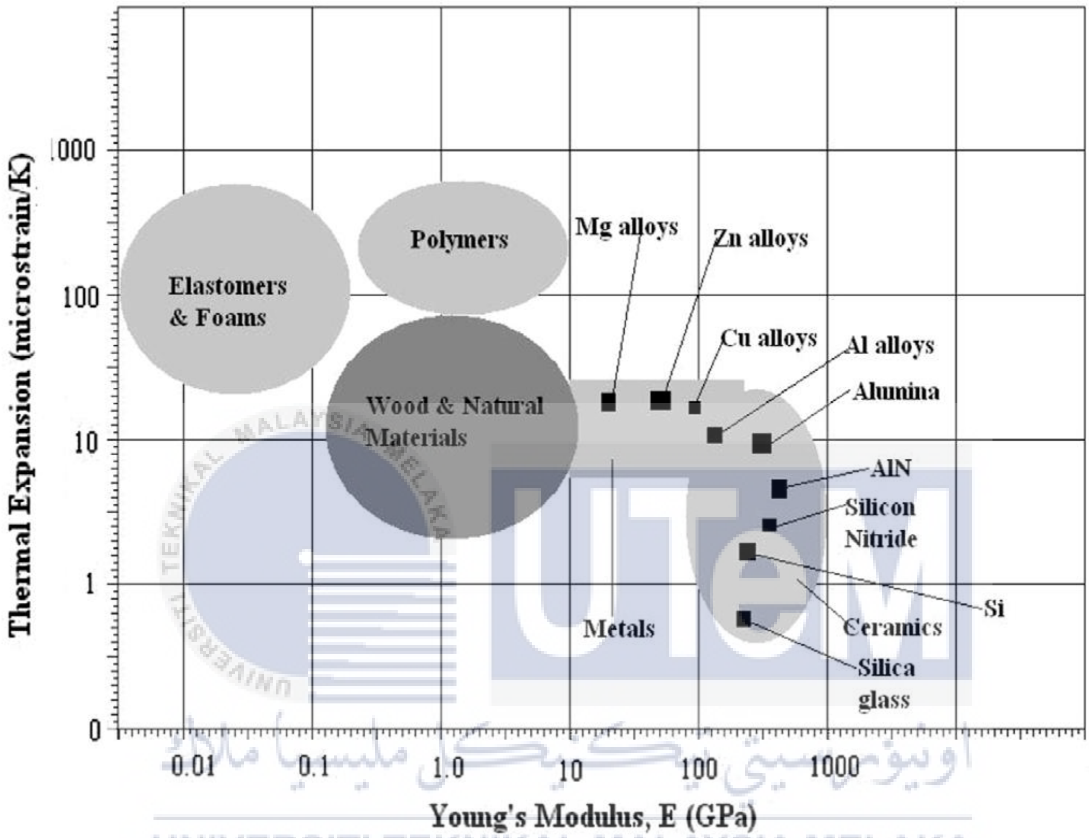


Figure 2.4 : Thermal expansion vs Young Modulus for different class of materials (G. Prashant Reddy & Navneet Gupta1, 2010)

Table 2.3 shows material properties of Aluminum Alloys that obtained from Metals Handbook (Rafael Nunes et al, 1990).

Table 2.3 : Mechanical properties of Aluminum Alloys

Mechanical Properties	Metric
Density	2.7 g/cc
Thermal Conductivity	167 W/m.K
Specific heat Capacity	0.896 J/g.ºC

2.2 Natural Convection

LED lighting has grown in the recent market. Nevertheless, Up to 70% of total energy consumed by LED lighting is emitted as heat which will affect the performance of LED lighting without a proper dissipation of heat. (Yu et al, 2011). Cooling of LEDs can be passive via natural convection or active including moving parts like fans. While the study of the active cooling system, moving parts were included which result in high manufacturing cost. With additional energy consumption, the complexity of the cooling system and regular maintenance needed. Therefore, the active cooling system only used for high powered LED lighting. Natural convection as in passive cooling is used for consumer LED lighting with the application of heat sink. (G. Schmid et al, 2017). There were several advantages of natural convection heat sink listed out by J.R. Pryde & D.C. Whalley, (2014) at 14th IEEE Intersociety Conference on Thermal and Thermomechanical Phenomena in Electronic Systems.

Advantages of Natural Convection Heat Sink

(1) Long Lifespan

(2) Zero power consumption

(3) Low to zero maintenance

(4) No noise

Under natural convection condition, the motion of fluid due to a body forces and their dependence on fluid density, which is sensitive to the temperature that accounted for the fluid (S. Mostafa Ghiaasiaan, 2011). According to U.V. Awasarmol & A.T. Pise,(2015), Natural convection from a heat sink used to simulate a huge variety of engineering applications and provide deep understanding into a more complex heat transfer system like a heatsink and etc.

There were several studies of natural convection of heat sink by a lot of researchers. (Yu et al, 2010) studies the natural convection around a radial heat sink, which in his study stated that the general flow pattern able to describe like a chimney. Which cooling air at outside entered the inner region of the heatsink and gain heated when passed through the fins of the heatsink. Besides, parametric studies of heat sink also were performed to study the effect of geometry toward thermal resistance and heat transfer coefficient. In the study stated there existed optimal values of geometry parameter for good performance of heat sink.

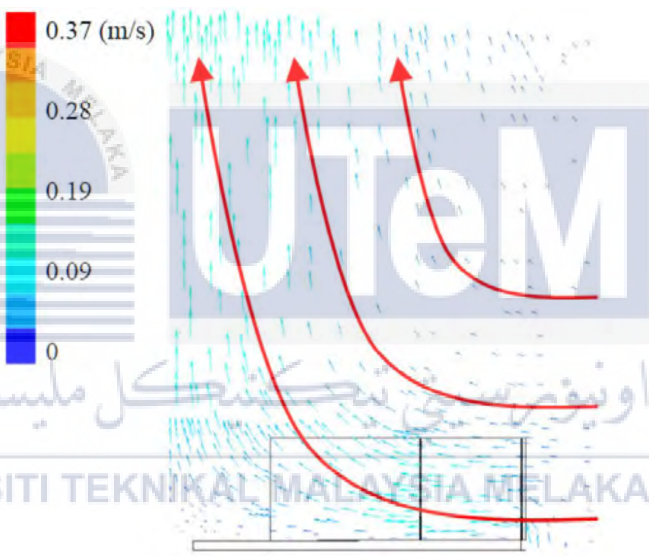


Figure 2.5 : General flow pattern of natural convection (S.J. Park et al,2015)

2.3 Geometric Optimization of Heat Sink

To optimize the heat sink performance, parametric studies usually performed to compare the effect of geometric parameters toward thermal resistance and heat transfer coefficient of the heatsink. There were studies to investigate natural convection heat transfer around a radial heat sink with a concentric ring performed by B. Li & C. Byon, (2015). The research studied the effect of fin length, the spacing between fins and Elenbaas number on the Nusselt number of the heatsink.

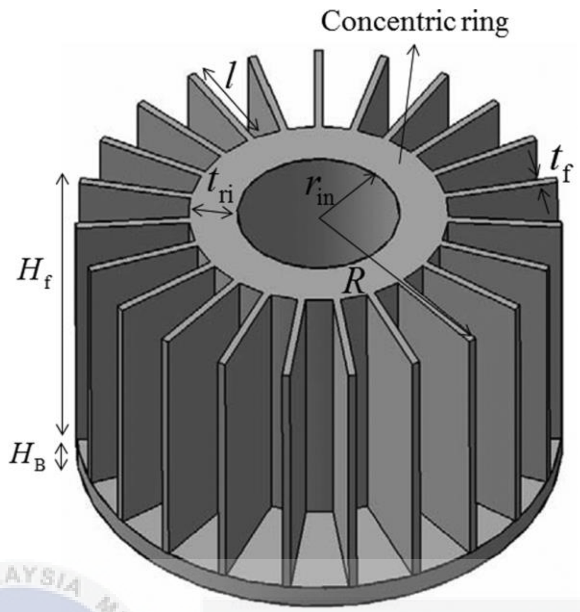


Figure 2.6 : Heat Sink with concentric ring (B. Li & C. Byon, 2015)

There was a study of natural convection heat transfer around radial heatsink with some perforated ring to enhance the thermal performance of heat sink with concentric ring. This model of heatsinks shown better thermal performance compared to the imperforated ring heat sink by 17% because unhindered free convective flow through the perforated ring. In the same time, this model successfully reduced the mass of heat sink by 37%. (B. Li et al, 2016)

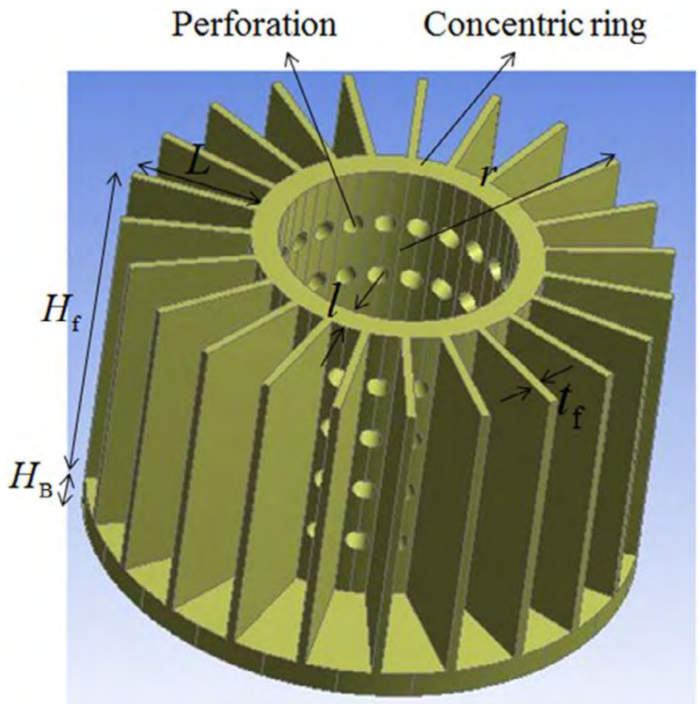


Figure 2.7 : Heatsink with perforation ring (B. Li et al, 2016)

In the studies of natural convection of radial heat sink performed by Yu et al, (2010), three geometric parameters had been studies which are fin length, fin height, and number of fins. Other studies by Yu et al, (2011), the study of type of heat sinks such as L model, LM model and LMS model were compared. The results are shown in figure 2.8a to 2.8c which the result of LM model considers as the most efficient model of heatsink. After selected the most efficient model, parametric studies were conducted with geometric parameters of number of fins, long fin length, middle fin length and heat flux toward thermal resistance and average heat transfer coefficient.

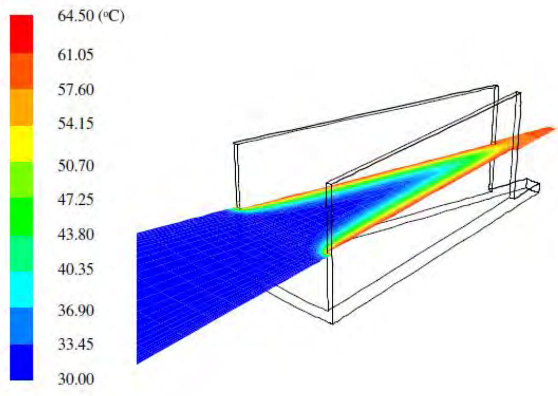


Figure 2.8a : L type model (T_{avg} = 61.96°C) (Yu et al, 2011)

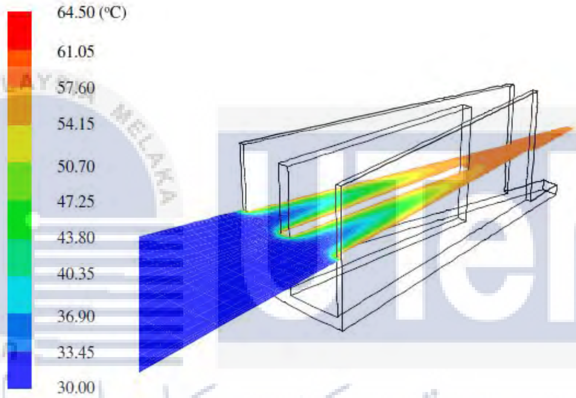


Figure 2.8b : LM type Model (T_{avg} = 57.96°C) (Yu et al, 2011)

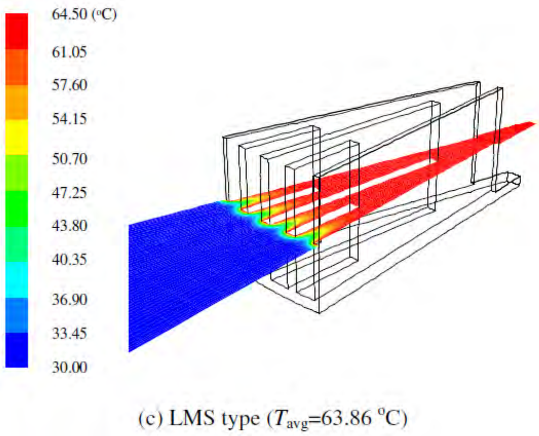


Figure 2.8c : LMS type Model (T_{avg} = 63.86°C) (Yu et al, 2011)

Since the focus is on consumer LED lighting. Therefore, optimization of LM type model will be studied in detail which the parametric studies will be focus on the height of fins and length of fins.

2.3.1 Geometric Parameter Height of Fins

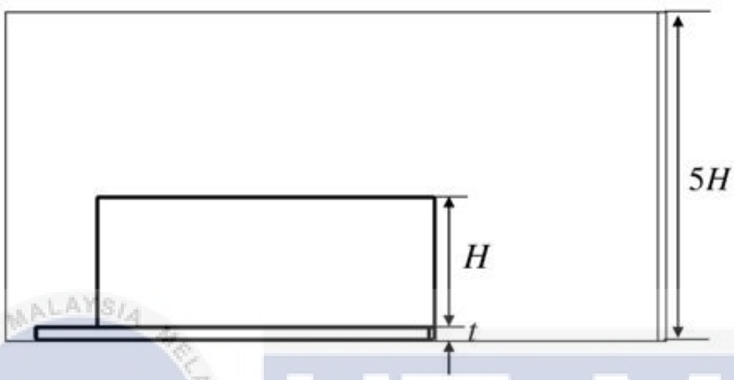


Figure 2.9 : Height of Fins (Yu et al, 2011)

For any given fin space, there exists an optimal fin height compare to a maximum rate of natural convection heat transfer. This statement was stated by Welling & Woolbridge, (1965) after performed an experimental study of vertical oriented rectangular fins of constant length attached to a vertical base. A numerical investigation conducted by Yu et al. (2010) to investigate natural convection of a radial heat sink indicated the effect of the fin height toward thermal resistance and heat transfer coefficient. The studies concluded that there was an improvement by an increment of fin height that caused by increasing in heat transfer surface area resulted in a lower thermal resistance heat sink. However, the change in heat transfer coefficient insignificant. This is because of small increment in velocity of entering air to the inner heat sink with increasing the fin height. Figure 2.10 shows the result of the numerical investigation.

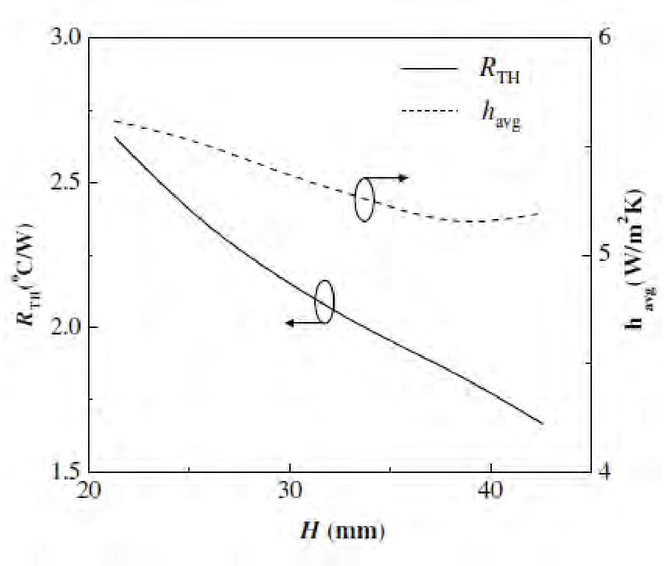


Figure 2.10 : The effect of the fin height (Yu et al, 2010)

Besides that, there is a study of the optimum design of heat sink related to fin height profile conducted by D. Jang et al. (2014). Fin height profiles proposed in the study which it able to improve the cooling performance without exceeding the mass that comparable to plate fin heat sink. There were three profiles that suggested in the study. Figure 2.11a to 2.11c show the configuration of the heat sink.

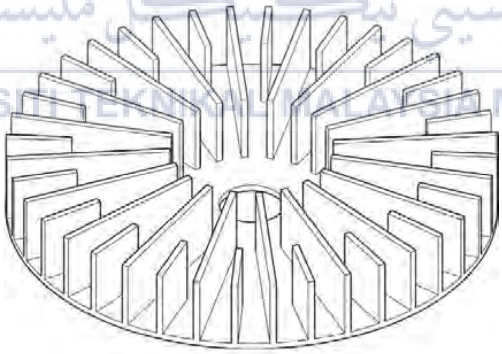


Figure 2.11a : LM plate-fin type (D. Jang et al, 2014)

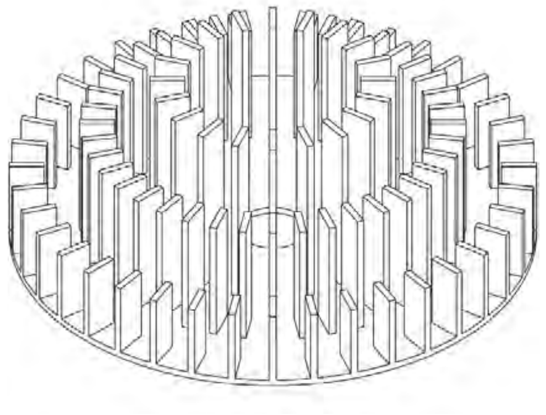


Figure 2.11b : Pin fin array with the tallest fins on the inside(Type 1)

(D. Jang et al, 2014)

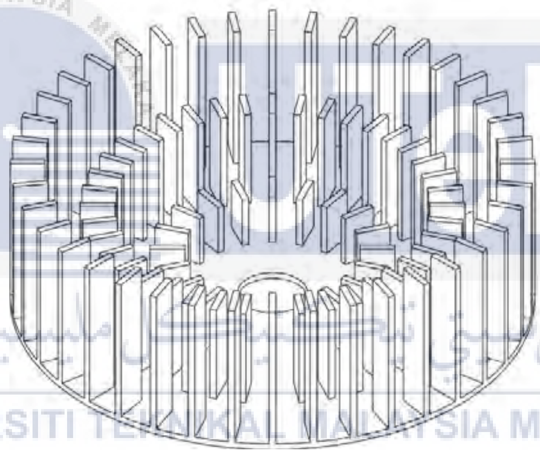


Figure 2.11c : Pin fin array with the tallest fins on the outside(type 2)

(D. Jang et al, 2014)

The result of his study as shown at figure 2.12.

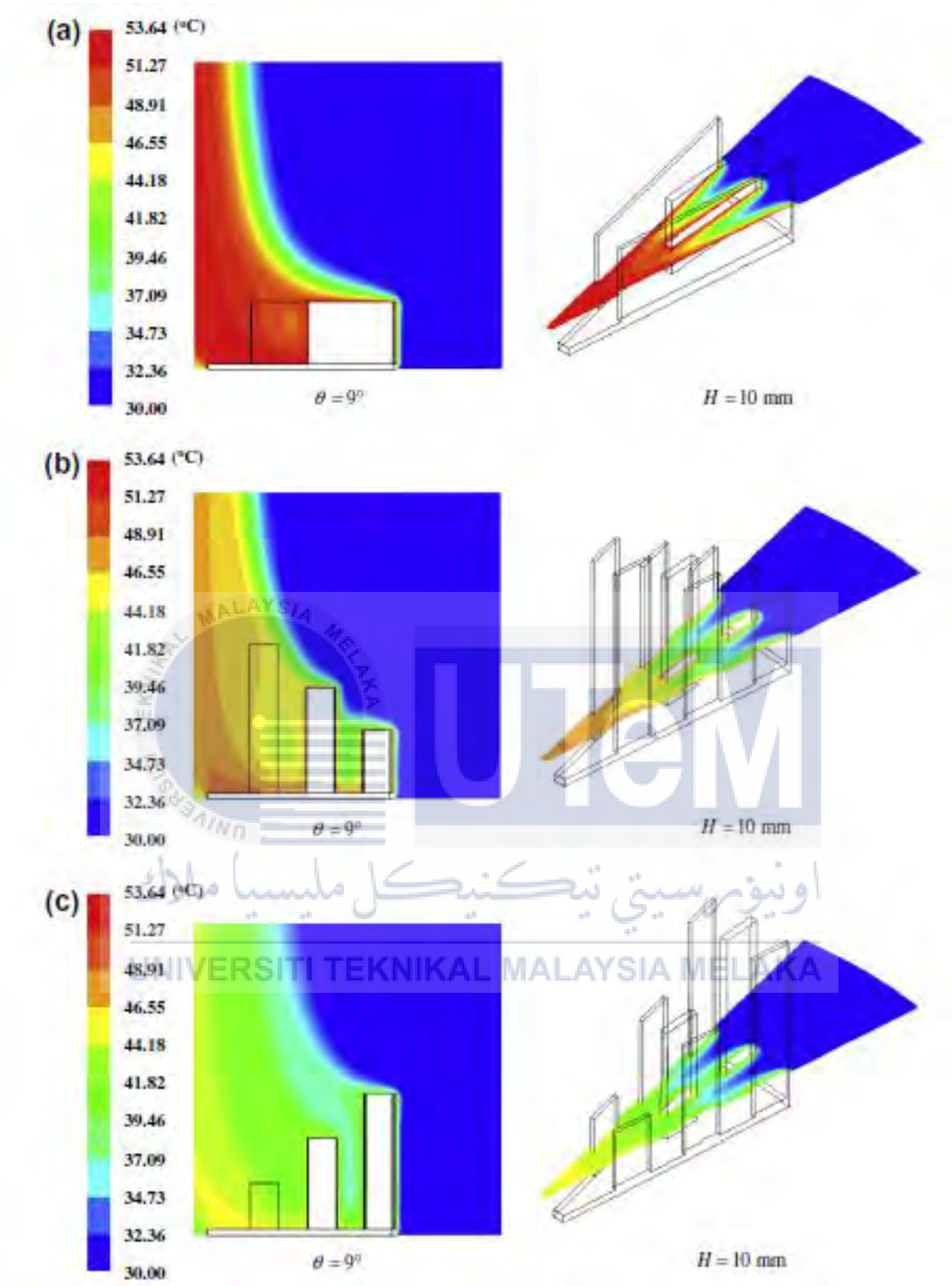


Figure 2.12 : Temperature contours at $\theta = 9^0$ and H = 10mm. (a) LM plate-fin model. (b) Pin fin array with the tallest fins on the inside. (c) Pin fin array with the tallest fins on the outside. (D. Jang et al, 2014)

Based on this study, the flow for different height of fin in heat sink will show chimney like pattern. Majority of the heat transfer occurs in the outer region and less heat transfer at the inner region of the heatsink which causes inflow rises. Thus, the cooling rate decreased in the inner region resulted in nonuniform heat transfer. The average temperature of the model of heatsink shown in table 2.4. Overall, the heat transfer of type 2 configuration shows the best performance of heat sink for different fin height profile. Therefore, the studies concluded that cooling performance of a pin-fin radial heat sink with a fin-height profile show huge improvement which more than 45% while did not increase the mass significantly.

Table 2.4 : Comparison of various fin height profile (D. Jang et al, 2014)

Model	A (mm^2)	L_O (mm)	L_D (mm)	$\dot{m}_{air}$ (10^{-5} kg/s)	T_{avg} (°C)
LM plate-fin	4499	21.3	0	3.04	53.12
Type 1	5000	22	15	4.39	48.86
Type 2	5000	46	15	6.4	44.80

In the research of V.A.F. Costa & A.M.G. Lopes (2014), he found out that by increasing the fin height, the thermal performance of heat sink also increased but with an increment of heat sink mass. The result of the study is shown in table 2.5.

Table 2.5 : Simulation result of changing fin height(V.A.F. Costa & A.M.G. Lopes, 2014)

Fin height H [m]	Core temperature T_c [°C]	Thermal resistance R [°C/W]	Heat sink mass M [kg]
0.021	73.5	2.005	0.197
0.025	68.0	1.719	0.223
0.030	63.6	1.49	0.256
0.035	59.6	1.281	0.289

2.3.2 Geometric Parameter Length of Fins

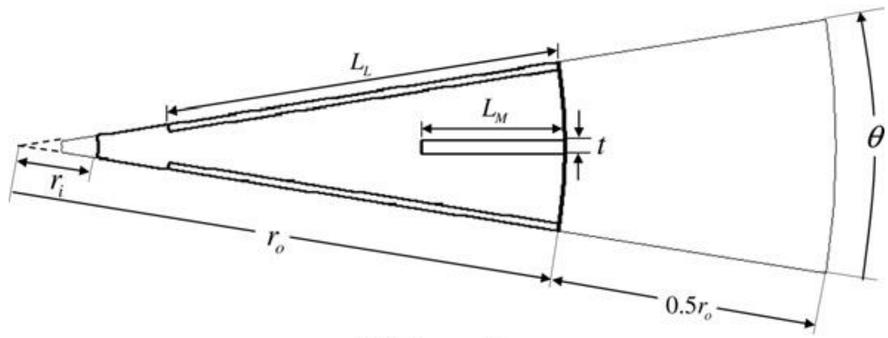


Figure 2.13 : Parameter of Length Fin (Yu et al,2011)

Majority of the study related to geometry optimization of heat sink involved an adjustment in length of fins. When Yu et al. (2010) study the natural convection around the heat sink, one of the parametric studies that done was changing the length of fins for the heatsink. The result of study for changing the length of fins show the positive result as the thermal resistance decreased but average heat transfer coefficient decreased. Thermal resistance reached a steady value when fin length reached a certain value because of heat sink temperature similar to the air temperature of the inner region. Therefore, the additional length of fin would not contribute to the heat transfer rate which similar to the statement claimed early. Existed optimal value for different parametric study. Figure 2.14 show the result of the study with the effect of fin length.

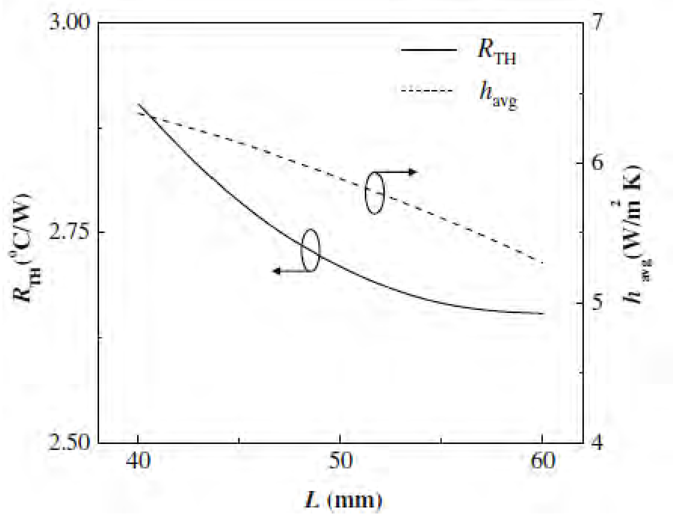


Figure 2.14 : Effect of the fin length (Yu et al, 2010)

Another study of optimization design of a radial heat sink under natural convection by Yu et al. (2011). The parametric study that the study did was changing the fin length of LM model. The optimization that done is changing the length of long fin with the fixed length of middle fin and change the length of middle fin with fix length of long fin. Figure 2.15 shows the result of changes in the length of long fin with the fixed length of middle fin effect on heat transfer coefficient, h and thermal resistance, R_{th}. Based on the figure, length of long fins increased, the thermal resistance shows decrease curve which means the cooling performance of heat sink increased. Thermal boundary layers between the next fins overlapped then fully developed as the flow of air proceed to the inner region. However, the thermal resistivity of heat sink increased after reached a certain length of increment which means for every optimization made.

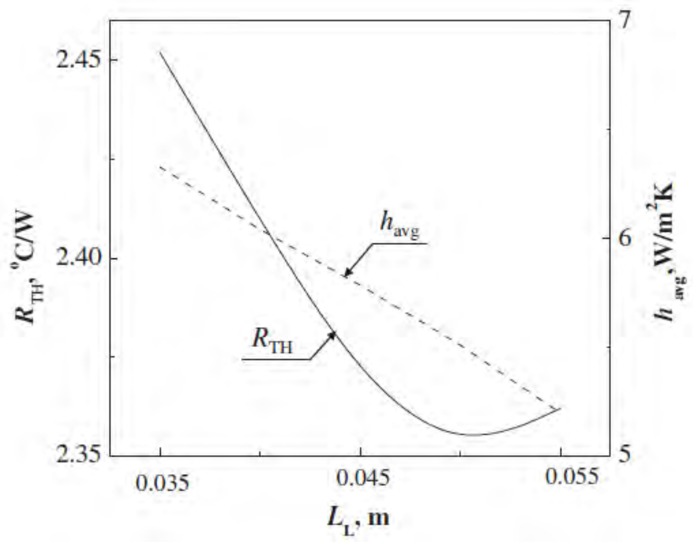


Figure 2.15 : Effect of Long Fin Length (Yu et al,2011)

Next will discuss the effect of middle fin length toward heat transfer coefficient, h and thermal resistance, R_{th}. The principle is similar to the adjustment length of long fin. The result is shown in figure 2.16. Small in length of middle fins, the thermal boundary layer did not affect long fins thermal boundary layer. Therefore, it shows good performance of the heat sink. However, as the length of middle fin increased, the thermal boundary layers of middle fin caused the thermal resistance decreased.

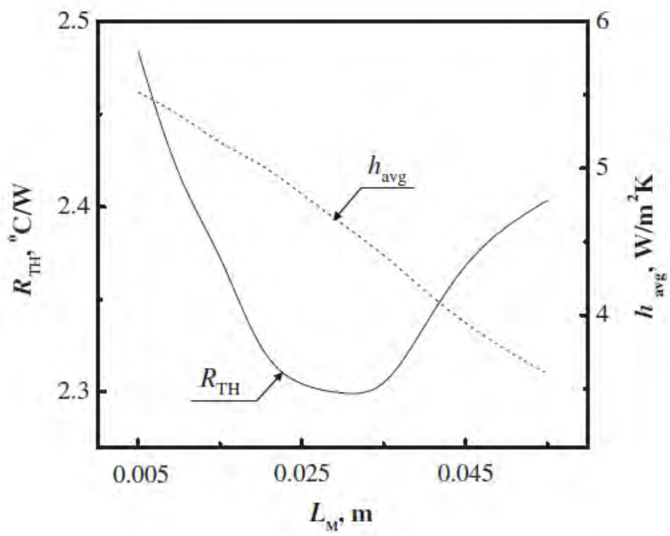


Figure 2.16 : The effect of middle fins length (Yu et al,2011)

2.3.3 Geometric Parameter Thickness of Fins

Other than the length of fin and height of fin for a heat sink parametric study, the thickness of fins also played an important role in optimization of the heatsink. According to the research done by Park et. al. (2016), the result toward the effect of the fin thickness as shown at figure 2.17. From the result, it pointed out that with the decrease in fin thickness, the performance was better as T_{max} of the heat sink dropped significantly. Heat Sink heat dissipation performance increased due to increase in the spacious gap between fins which allowed more convective heat transfer.

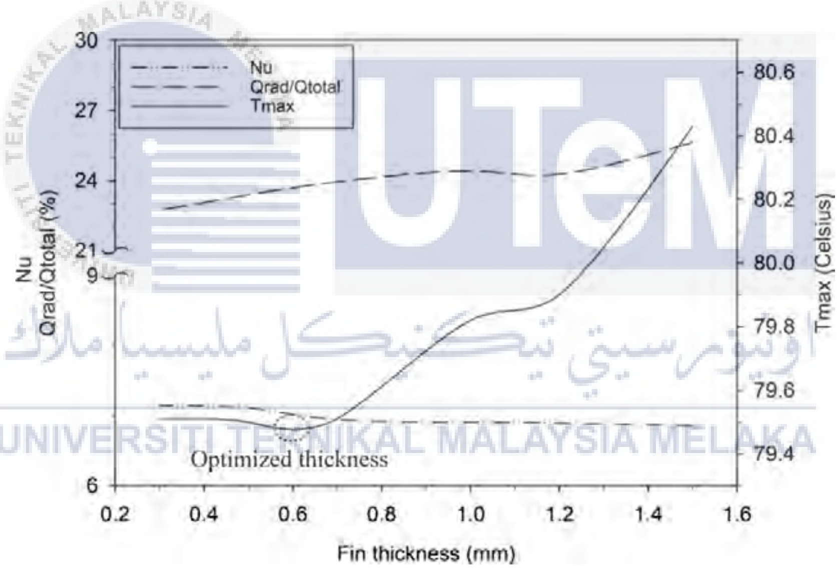


Figure 2.17 : The effect of the fin thickness (Park et. al., 2016)

2.4 Numerical methods (Simulation)

Numerical methods to simulate the result of experimental were widely used. This is because the differences between experimental result and the numerical result were small. Although there were differences between the numerical result and experimental result, the difference is less than 6.1% which the simulation able to accurately simulate the studies. (D. Jang et al, 2011).

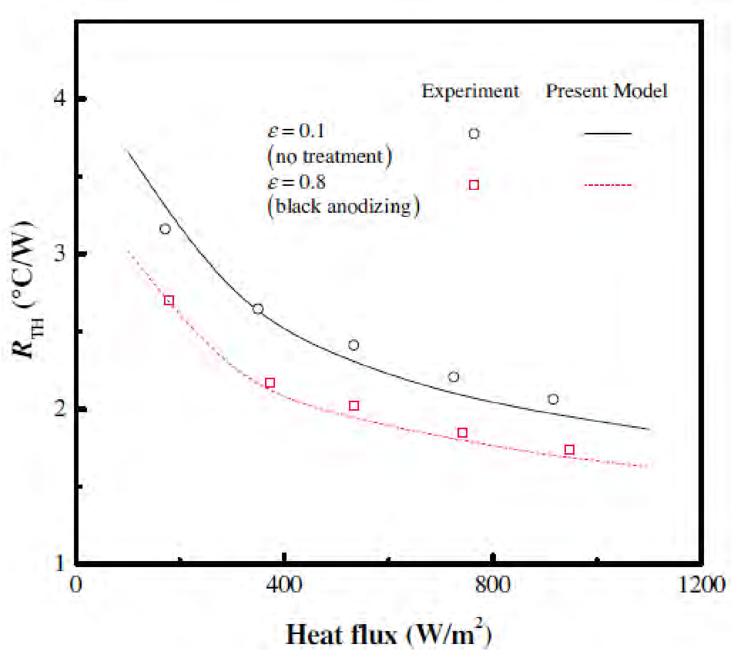


Figure 2.18 : Comparison of experiment result and numerical result (D. Jang et al., 2011)

CHAPTER 3

METHODOLOGY

3.1 Introduction

In this chapter, description of the methodology used in the project will be done. The flow of project will be summarized in a flowchart as shown in figure 3.1. It started with literature review to understand the theory of cases that will be studied. The literature review mainly to understand the effect of parameters changes for heatsink which will be useful for analyzing the results later.

Next, will be model setup. The model setup included geometry drawing, meshing and simulation. This step was important to determine the suitable methods to run the numerical analysis. After this step, validation was done. Validation was done to compare the closeness of numerical result toward an experimental result.

After validation, simulation with case study was done. The steps were to run a simulation with other cases that needed to be studied. After case studies were done, analysis of the result was done. Analysis of the result was to determine which geometric parameters affect the performance of heat sink the most without changing of mass. After solution proposed, report writing were done.

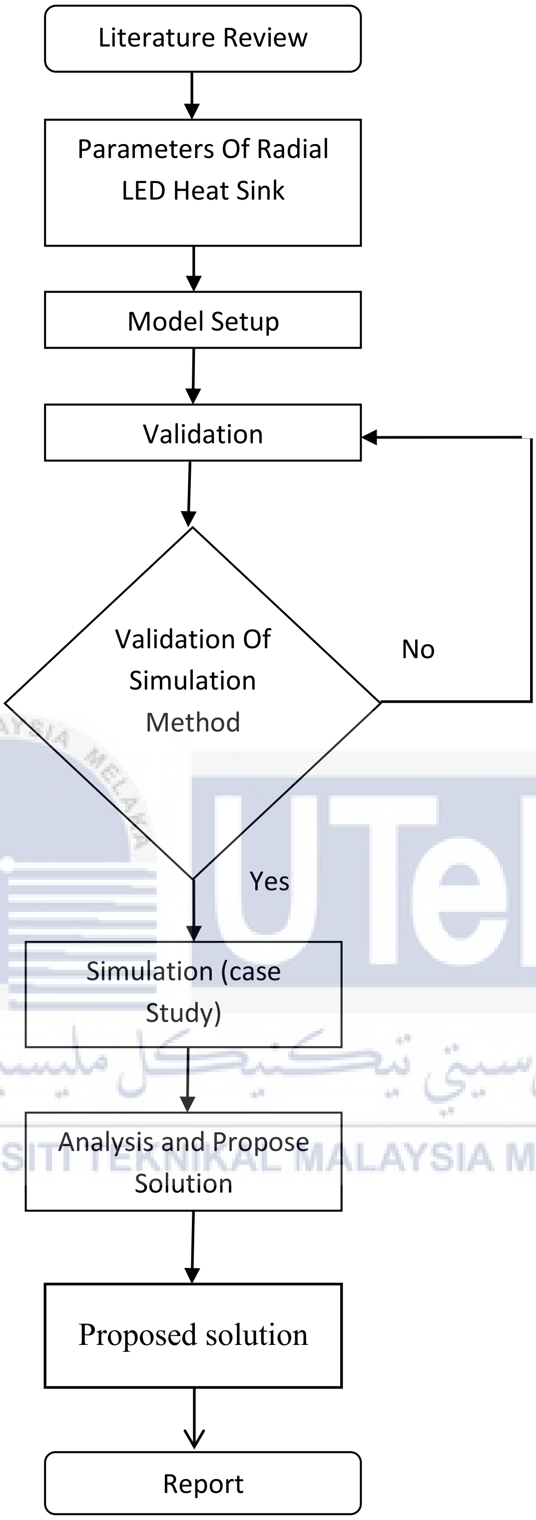


Figure 3.1: Flowchart of the methodology.

3.2 Geometry Drawing

In this section, the geometry parameters of the reference heat sink must be followed to be validated. The geometry of heat sink as shown in figure 3.2 and table 3.1.

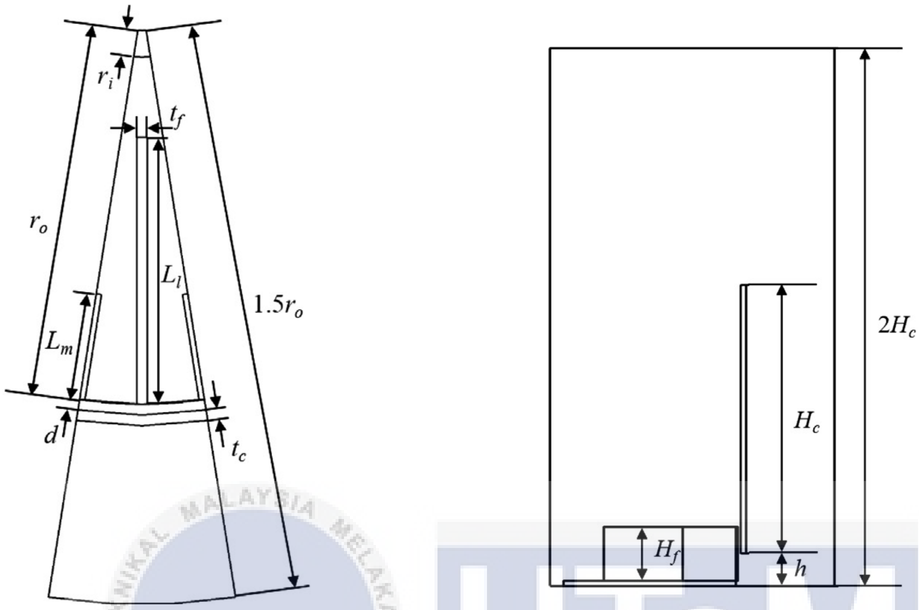


Figure 3.2 : (a) Side View and (b) Top View of heat sink along with hollow cylinder and computational analysis domain (S.J. Park et al,2015)

Table 3.1 : Geometry parameters of the reference heatsink (S.J. Park et al, 2015)

Parameters	Value
Number of Fin, N	20
Long Fin length, L_l	40 mm
Middle Fin length, L_m	20 mm
Height of Fin, H_f	20 mm
Thickness of fin,t_f	2 mm
Outer radius, r_o	75 mm
Inner radius, r_i	5 mm
Radius of computational analysis domain	1.5 r_o
Height of computational analysis domain	2 H_c

The radius and height of computational analysis domain are set as 1.5 r_o and 2H_c is because the average temperature change of numerical result was less than 1%. (S.J.

Park et al, 2015). The computational analysis domain is important to show the result of fluid flow around the heat sink.

3.3 Parameters Of Optimization

In order to proceed to geometric optimization of LED heat sink, parametric study needed to be done. Parametric studies related to natural convection of radial heat sink to fully understand the working principle and theory if there were changes of parametric toward the effect on heat transfer of heat sink. By study the journal which related to optimization design of heat sink, the category of parametric can be finalized at the list below.

Parametric study of natural convection heat sink:

a) Height of Fins of Heat Sink
b) Length of Fins of Heat Sink
c) Number of Fins of a Heat Sink
d) Thickness of Fins of a Heat Sink
e) Material of a Heat Sink
f) Orientation of Heat Sink
g) Number of Perforated Ring of Heat Sink

In this study, the height of fins of heatsink, length of fins and thickness of heat sink will be focused to fulfill the objective of the title. Either increase or decrease in height of fins, length of fins and thickness of heat sink, the mass of heat sink should remain or lower than the original.

3.4 Boundary Conditions

The boundary conditions of the fluent setting were shown in figure 3.6. The heat flux that applied at the bottom of the heat sink was 1000 W/m^2. The wall at the side was set as an interface so that able to set as a periodic boundary condition. The top of the domain was set as air outlet while there were two inlet of air.

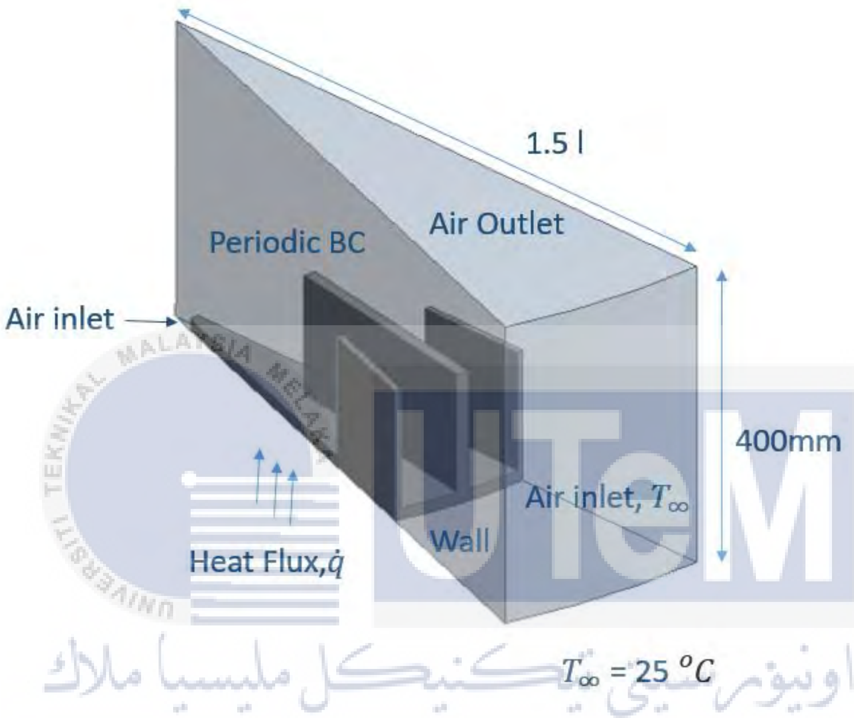


Figure 3.3 : Boundary condition of heatsink

3.5 Physical Properties

Table 3.2 showed the physical properties of heat sink and air used for case study. The setting at Ansys fluent for materials will be set as below.

Table 3.2 : Properties of Air and Aluminium

Properties	**Air**	**Aluminium**
Density(kg/m^3)	Ideal-gas	2719
Thermal conductivity (W/m.K)	0.0242	171
Specific heat (J/Kg.K)	1006.43	871
Viscosity (Kg/m.s)	1.7894 x 10^{-05}	-

3.6 Simulation Software

The simulation proceeds by using Ansys Ver 16.0 Software which the software include Geometry, Meshing and Fluent. Ansys is a popular software that used for simulate numerical result of the certain condition not only on fluids but also structure and others. The simulation of geometry optimization of natural convection heat sink will do using Ansys. Computational Fluid Dynamics (CFD) is a branch of Ansys under Fluent which used to solve and analyze the fluid flow pattern using numerical method. The advantages of using numerical methods are to save the cost of building a prototype for studies or even running the experiment for every case at wind tunnel.

3.7 Meshing

For a better result of numerical, meshing played an important role. To validate the experimental result, the tetrahedral mesh was used. The meshing of the heat sink separated into three section as shown in figure 3.4.

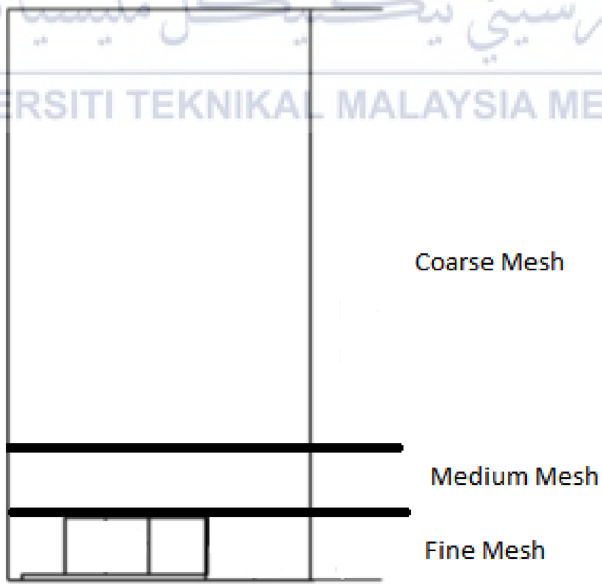


Figure 3.4 Three section of different quality mesh of heat sink

The meshing of computational analysis domain and heat sink separated into three section is because the finer mesh around the heat sink will show the more accurate result of fluid flow pattern. The reason for the second section is medium mesh because the flow pattern of leaving the inner region of heatsink is important to study as in the height of fins will be increased for further study. Coarse mesh for the last section because the fluid flow of the last section did not affect the numerical result of heat sink significantly compared to the first section.

3.8 Governing Equations

The governing equations are as follows:

Air Side:

Continuity equation:

$$\frac{\partial(\rho u)}{\partial x}+\frac{\partial(\rho v)}{\partial y}+\frac{\partial(\rho w)}{\partial z}=0 \tag{3.1}$$

Where ρ is density(kg/m^3), u is x-component of velocity (m/s), v is y-component of velocity (m/s) and w is z-component of velocity (m/s)

Momentum equations:

$$\frac{\partial(\rho u^2)}{\partial x}+\frac{\partial(\rho uv)}{\partial y}+\frac{\partial(\rho uw)}{\partial z}= -\frac{\partial P}{\partial x}+\mu\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}+\frac{\partial^2 u}{\partial z^2}\right)$$

$$\frac{\partial(\rho vu)}{\partial x}+\frac{\partial(\rho v^2)}{\partial y}+\frac{\partial(\rho vw)}{\partial z}= -\frac{\partial P}{\partial y}+\mu\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}+\frac{\partial^2 v}{\partial z^2}\right)+g(\rho-\rho_a) \tag{3.2}$$

$$\frac{\partial(\rho wu)}{\partial x}+\frac{\partial(\rho wv)}{\partial y}+\frac{\partial(\rho w^2)}{\partial z}= -\frac{\partial P}{\partial x}+\mu\left(\frac{\partial^2 w}{\partial x^2}+\frac{\partial^2 w}{\partial y^2}+\frac{\partial^2 w}{\partial z^2}\right)$$

Where ρ is density (kg/m^3), P is pressure (Pa), u is x-component of velocity (m/s), v is y-component of velocity (m/s), w is z-component of velocity (m/s), g is gravity and μ is dynamic viscosity, N/m^2s.

Energy equation:

$$\frac{\partial(\rho uT)}{\partial x}+\frac{\partial(\rho vT)}{\partial y}+\frac{\partial(\rho wT)}{\partial z}=\frac{k}{C_p}\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2}\right) \tag{3.3}$$

Where ρ is density (kg/m^3), u is x-component of velocity (m/s), v is y-component of velocity (m/s), w is z-component of velocity (m/s), T is temperature in ^{o}C and Cp is coefficient of heat capacity, J/(kg^{o}C)

Solid side:

Energy equation:

$$\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2}\right)=0 \tag{3.4}$$

Where T is the temperature in ^{o}C

3.9 Fluent or Solver (Computational Fluid Dynamics)

Fluent was used for numerical analysis. To proceed to the solver efficiently, the single period domain of the heat sink was calculated with periodic conditions as shown in figure 3.3. Besides, several assumptions were done to simulate the heat sink under natural convection.

Assumptions made:

a) Flow is steady and three dimensional

b) Properties of air are constant, except density

c) Ideal gas is used as air

d) Radiative heat transfer negligible at the heat sink

Semi-implicit method for pressure linked equations will be used to calculate the flow field with coupled pressure and velocity. In order to achieve a better result, convective terms of governing equations were discretized via a second-order upwind scheme.

When proceeding solver, there appeared difficulty for the continuity equation to converge. In order for the solver to converge, setting at under-relaxation factor needed to be changed. The initial fluent setting at under-relaxation factors was shown in table 3.3. The setting was changed for every 100 iterations that run through calculation. The changed setting was pressure increased by 0.1 while momentum decreased by 0.1. The summation of pressure and momentum would be 1.0. These processes repeated until the solution was converged.

Table 3.3 : Initial setting for under-relaxation factors

Elements	**Values**
Pressure	0.1
Density	1
Body Forces	1
Momentum	0.9
Turbulent Kinetic Energy	0.8
Turbulent Dissipation Rate	0.8
Turbulent Viscosity	1
Energy	1

3.10 Verification

In this section, there will be two test to be done. First is grid dependency test to determine the effect of different mesh size toward thermal resistance. Second, comparison of turbulence models will be done. This is to determine which turbulence model suitable to perform validation and case study.

3.10.1 Grid Dependency Test

In this section, different mesh size will be tested. Different mesh size will produce a different numerical result. In order to justify which mesh size should be used, three different mesh size were analyzed. Figure 3.5 showed the effect of thermal resistivity with different mesh size. The percentages error of 800,000 mesh size show 12.16%, 1 million of mesh size show 6.09% of percentages error while 1.25 million of mesh size show error of 4.97%. This show that with an increase of mesh size, the result differ. When mesh size increased more than 1 million, the variation of percentages error lowered. This indicated percentages error of medium grid case and fine grid case were very close. Thus, a heat sink with a mesh size of 1 million was selected to run the simulation.

Table 3.4 : Grid Dependency Test

Grid Type	**Coarse**	**Medium**	**Fine**
Mesh Size	809,870	1,038,569	1,252,071
Thermal Resistance, Rth (°C/W)	2.47876	2.34462	2.31983
Percentages Error, %	12.16	6.09	4.97

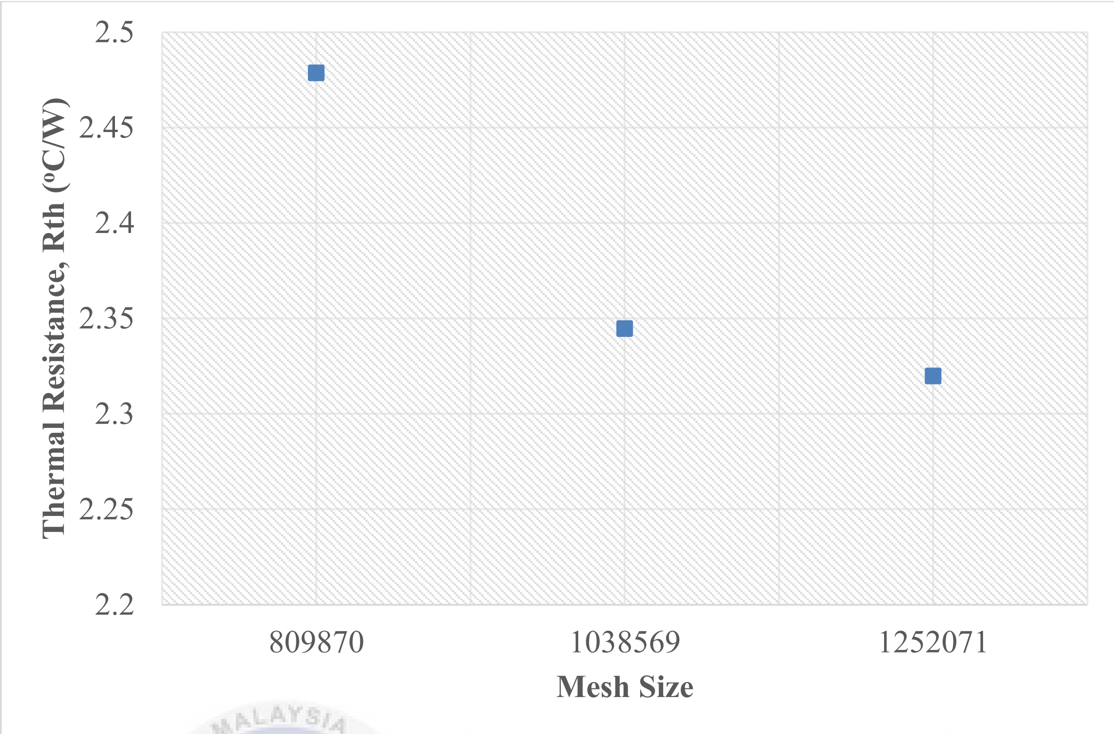


Figure 3.5 : Effect of mesh size toward thermal resistance

3.10.2 Comparison of Turbulence Models

After grid dependency test, comparison of turbulence models in the fluent setting was done. By changing the model, the turbulence model that produce accurate result were obtained. Turbulence model RNG k-ε produced the least percentages error among the turbulence model tested. Therefore, RNG k-ε turbulence model was selected for a case study. Table 4.1 shown the data obtained from the numerical analysis.

Table 3.5 : Thermal resistivity of different turbulence model

Turbulence Model	Thermal Resistivity, R_{th} (ºC/W)	Percentages error, %
RNG k-ε	2.34462	6.09
Standard k-ε	2.34473	6.10
Realizable k-ε	2.34515	6.12

3.11 Validation

This section will discuss the result of comparison between experimental result and numerical result. Figure 4.1 showed the difference of the results. The numerical result had the similar trend as an experimental result although numerical results did not fall into the range of experimental result. Therefore, this indicated that the method used for fluent setting was acceptable.

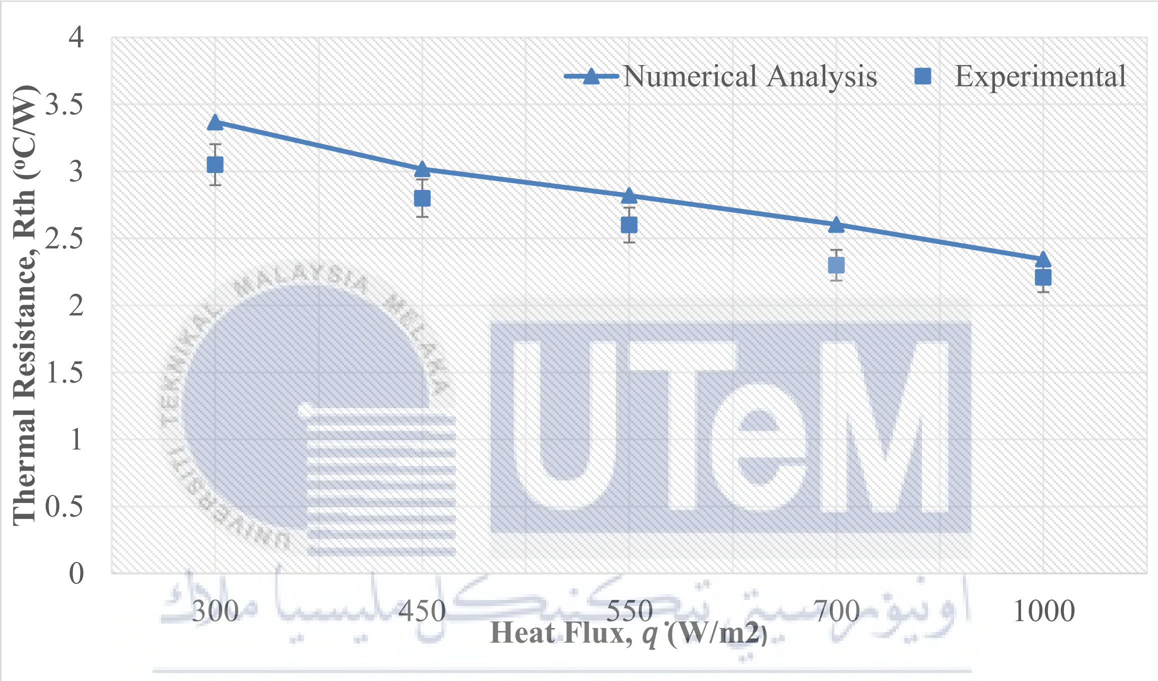


Figure 3.6 : Validation result

3.12 Simulation with Case Study

After validation is done, repeat the same method used in validation of solver by changing the ratio of the height of fin and length of the fin. In the same time, the ratio of the mass of heat sink to mass of reference heatsink should not be more than one. The solution from solver is saved, graphs of result should be generated include:

a) Comparison of Rth/Rth ref versus height of fin

b) Comparison of Rth/Rth ref versus length of fin

c) Comparison of Rth/Rth ref versus ratio of length of fin/height of fin

d) Comparison of Rth/Rth ref versus ratio of height of fin/thickness of fin

Thermal Resistance of heat sink can be defined as follows:

$$R_{TH} = \frac{T_{ave\_heatsink} - T_{\infty}}{\dot{q}_{heatsink\_base} \cdot A_{heatsink\_base}}$$

CHAPTER 4

RESULT AND DISCUSSION

4.1 Introduction

In this section, the result of simulation will be discussed. This included the changes of parameters toward the performance of heat sink. The parameters that studied included height of fin, length of fin and thickness of fin. The geometric optimization of heat sink would be done without increasing in mass to preserve the cost of material.

4.2 Geometric Parameter : Height Of Fin

In this section, the effect of decreasing in height of fin toward thermal resistance, R_{th} will be discussed. From the result, thermal resistance increased when the height of fin decreased was shown in figure 4.1. This indicated that the performance of the heat sink reduced when the height of fin decreased. This caused by increased in the heat transfer area which enables the flow of air from the outer region process with heat transfer when moving into the inner region. Figure 4.2 a and b showed the temperature of the heatsink, 20 mm height of fin showed the lowest temperature of the heat sink while the highest for 10mm height of fin configuration. Table 4.1 showed the increased by 93.33% in the surface area led to thermal resistance decreased. Therefore, the performance of the heat sink increased.

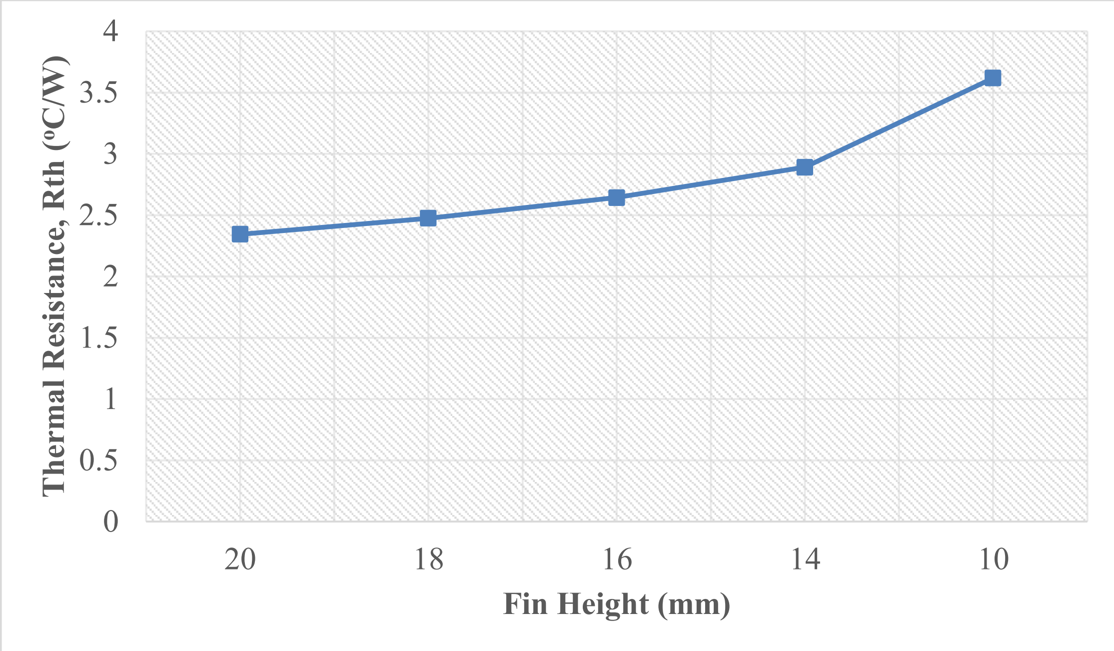
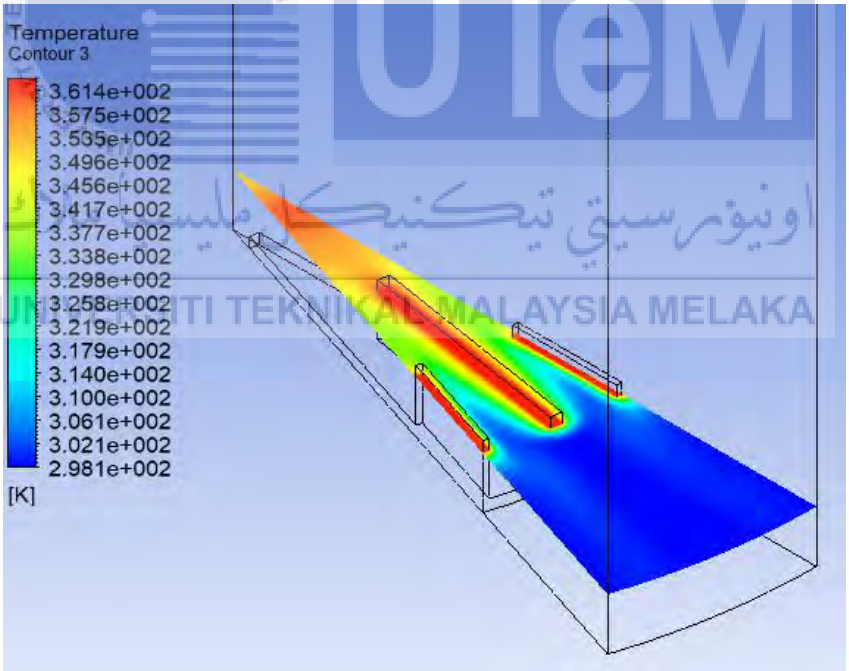
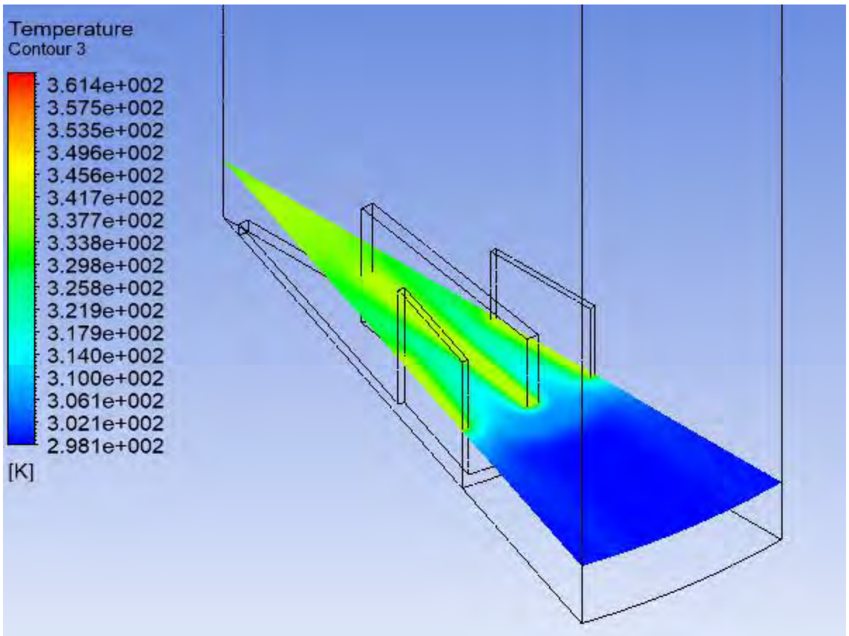


Figure 4.1 : Effect of fin height toward thermal resistivity



a) Height of fin 10mm

Figure 4.2 : Temperature contour of heat sink at y = 8mm



b) Height of fin 20mm

Figure 4.2 : Temperature contour of heat sink at y = 8mm

Table 4.1 : Effect of height of fin

Height (mm)	Thermal resistance, Rth (°C/W)	Surface Area (mm^2)
20	2.34462	3480
10	3.61785	1800

4.3 Geometric Parameter : Length Of Fin

In this section, the effect of length of fin toward thermal resistance will be discussed. Although there was two configurations for LM type heat sink which was long fin length and middle fin length. The effect of thermal resistance with both configurations was similar according to research done by yu et. al. (2011). Therefore, only long fin length will be simulated to check if the result similar with Yu et. al. (2011) theory. The numerical result has the similar trend with Yu et. al. (2011) which proved that the theory was correct.

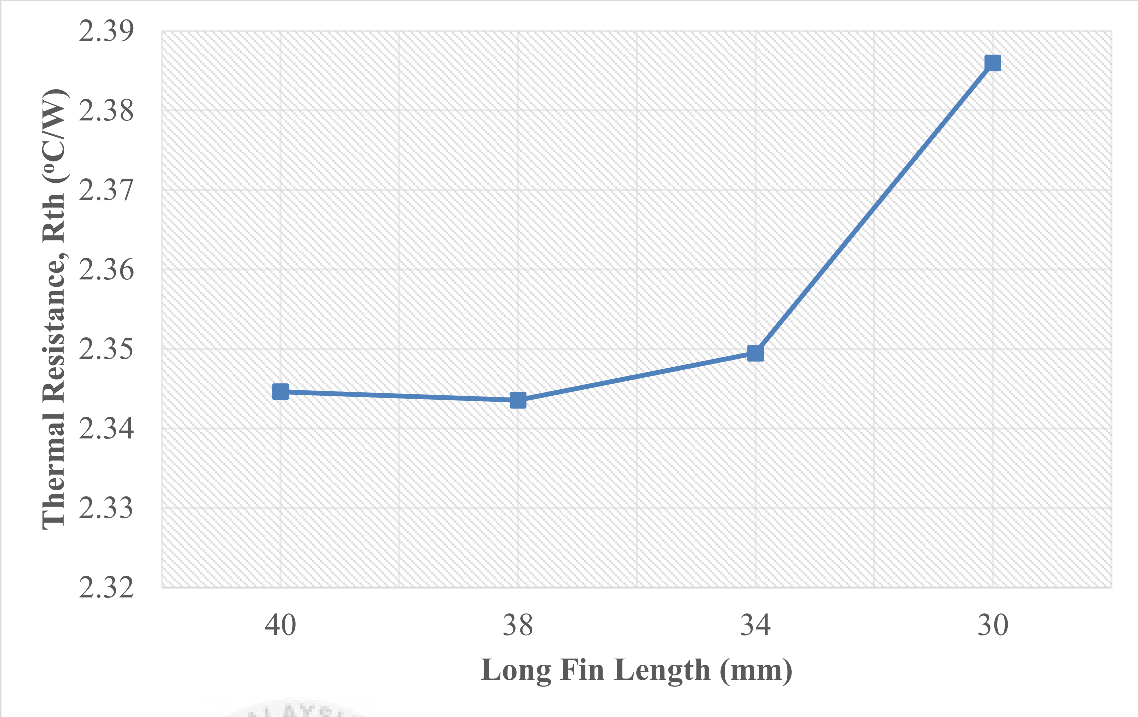
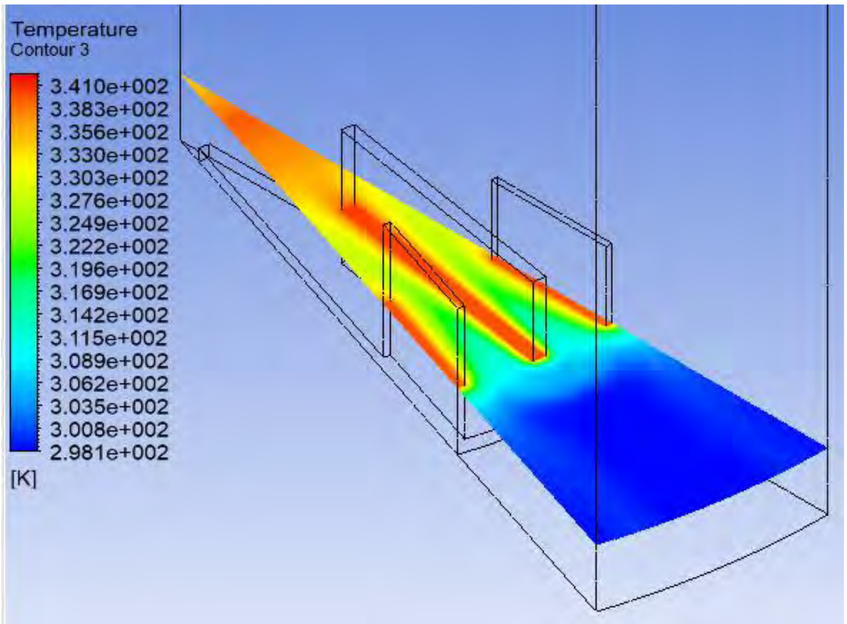
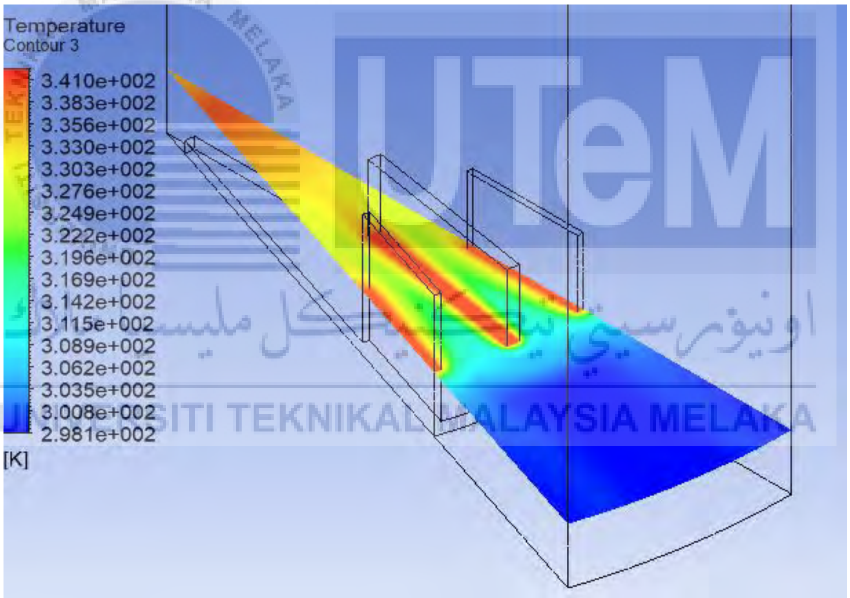


Figure 4.3 : Effect of long fin length toward thermal resistance

When the length of long fins increased, decreased in thermal resistance curve which meant the cooling performance increased. Cooling performance increased because thermal boundary layers between the next fins overlapped then developed as the flow of air proceed into the inner region. While for the middle fin length, the concept will be similar. When increase in middle length, thermal boundary layer formed did not affect the long fins thermal boundary layer. Other than the development of thermal boundary layers among fins, the surface area of heat sink fin increased as the length of long fin length increased. This meant the heat transfer area also increased. Therefore, the cooling performance of heat sink will be increased. The boundary layer formed was shown in figure 4.4.



a) Long fin length of 40mm



b) long fin length of 30mm

Figure 4.4 : Temperature contour of heat sink at y = 10mm

Table 4.2 : Effect of length of fin

Length (mm)	Thermal resistance, Rth (°C/W)	Surface Area (mm^2)
40	2.34462	3480
30	2.38598	3060

4.4 Case Study 1 : Thermal Resistance Versus Height of Fin With Fixed Mass

In this section, the effect of increase or decrease in height of fin with fixed mass toward thermal resistance will be discussed. Increase in height of fin will decrease in length of fin. This process was to fix the mass so that material cost would not increase. Table 4.3 showed the result of numerical analysis for the case study 1. The ratio of height/ long length was showed in the table below.

Table 4.3 : Numerical analysis of case study 1

No	Height (mm)	Length of fin(mm) long / middle	Ratio of Height / Length	Mass ratio	Thermal Resistance , R_{th}	Increment percentage , %	Surface Area, (mm^2)
1	16	50/25	0.3200	1.0000	2.5947	-10.67	3478
2	20	40/20	0.5000	4159.4mm^3 **reference**	2.3446	**reference**	3480
3	25	32/16	0.7813	1.0000	2.1843	6.84	3496
4	32	25/12.5	1.2800	1.0000	2.1132	9.87	3531
5	40	20/10	2.0000	1.0000	2.0097	14.28	3580
6	50	16/8	3.1250	1.0000	1.9291	17.72	3648

Figure 4.5 showed the data obtained from the numerical analysis. From the graph, as the height of fin increased, the length of fin for both middle fin and long fin reduced. Thus, the thermal resistance reduced. This indicated the cooling performance of the heat sink increased.

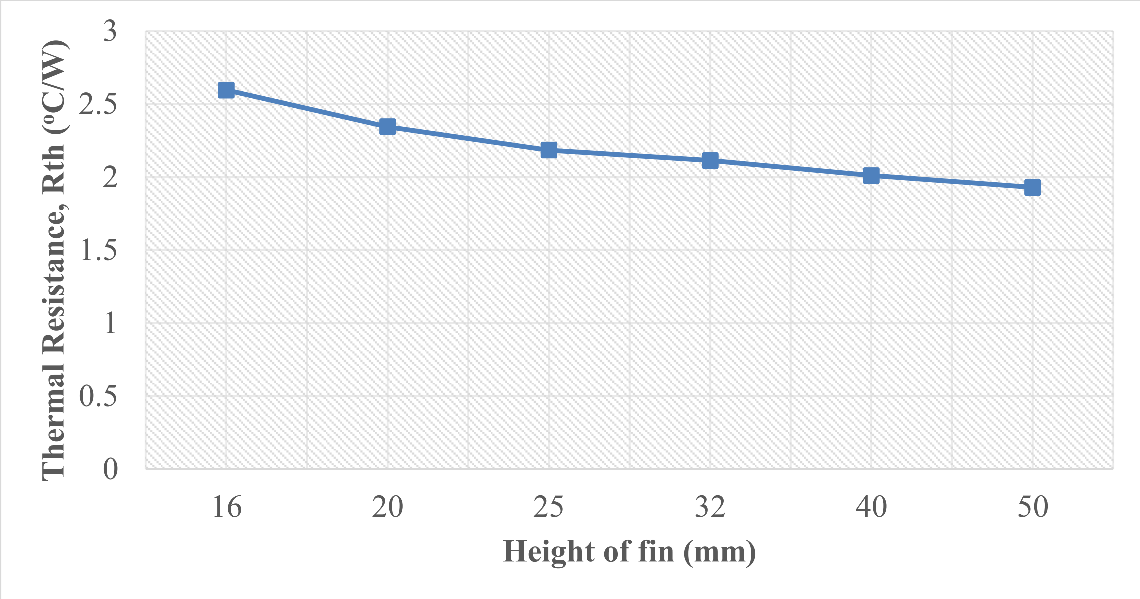
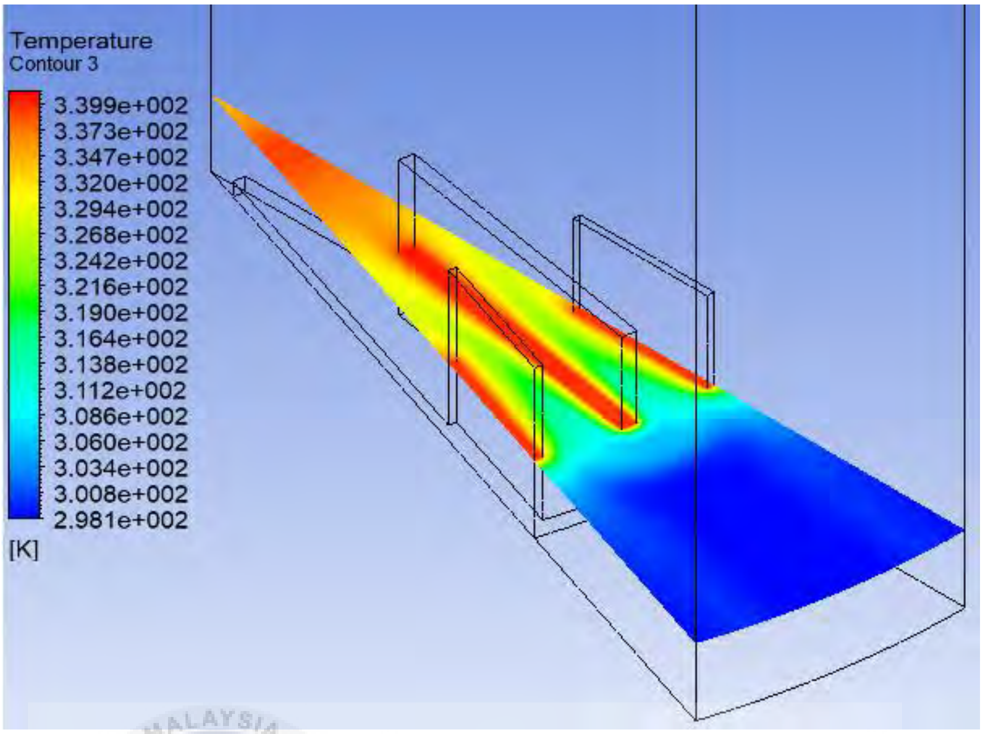
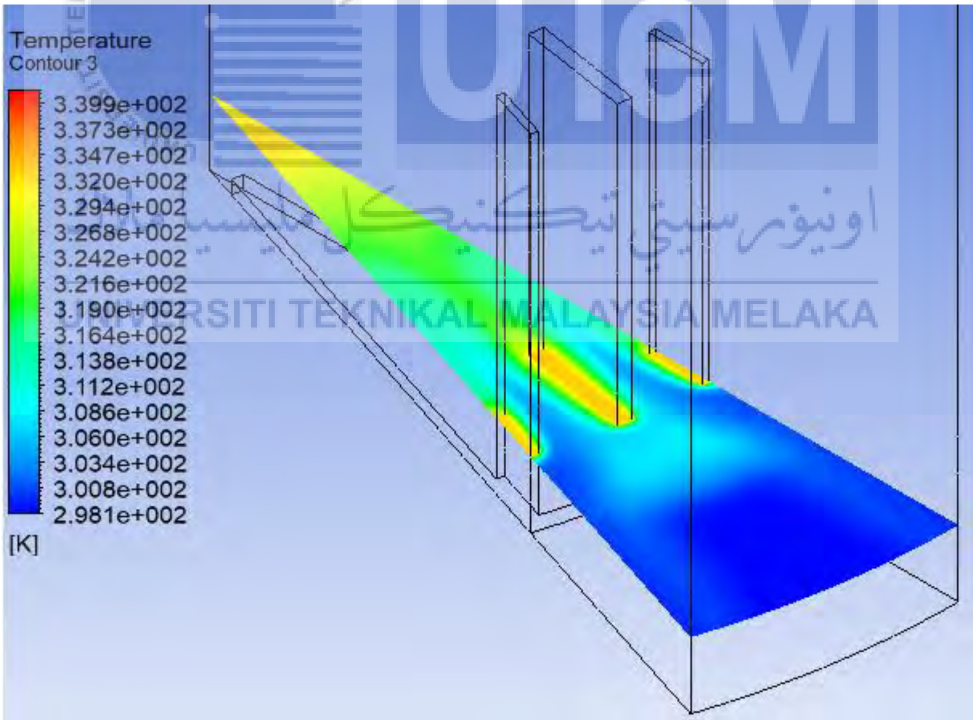


Figure 4.5 : Effect of height of fin toward thermal resistance

Based on figure 4.6, temperature contour of the heatsink was shown. Formation of the thermal boundary layer can be seen, the case a in figure 4.6 developed fully which enabled the outer region of air to enter to the inner region but for case b, the development of thermal boundary was too early which caused the outer region did not enter to the inner region efficiently. Due to fixed mass were considered for the case study, decreased in the length of fin will result in the height of fin increased. Based on table 4.3, the height of fin increased, the heat transfer area of the heat sink also increased by 4.83% for case b compared to case a which resulted in a better efficiency of cooling performance of heat sink. The result of case study 1 had shown that increase in height of fin will bring better improvement toward cooling performance of heat sink instead of increasing the length of the heatsink. Figure 4.8 showed when increasing in surface area, the cooling performance of heatsink will be improved.

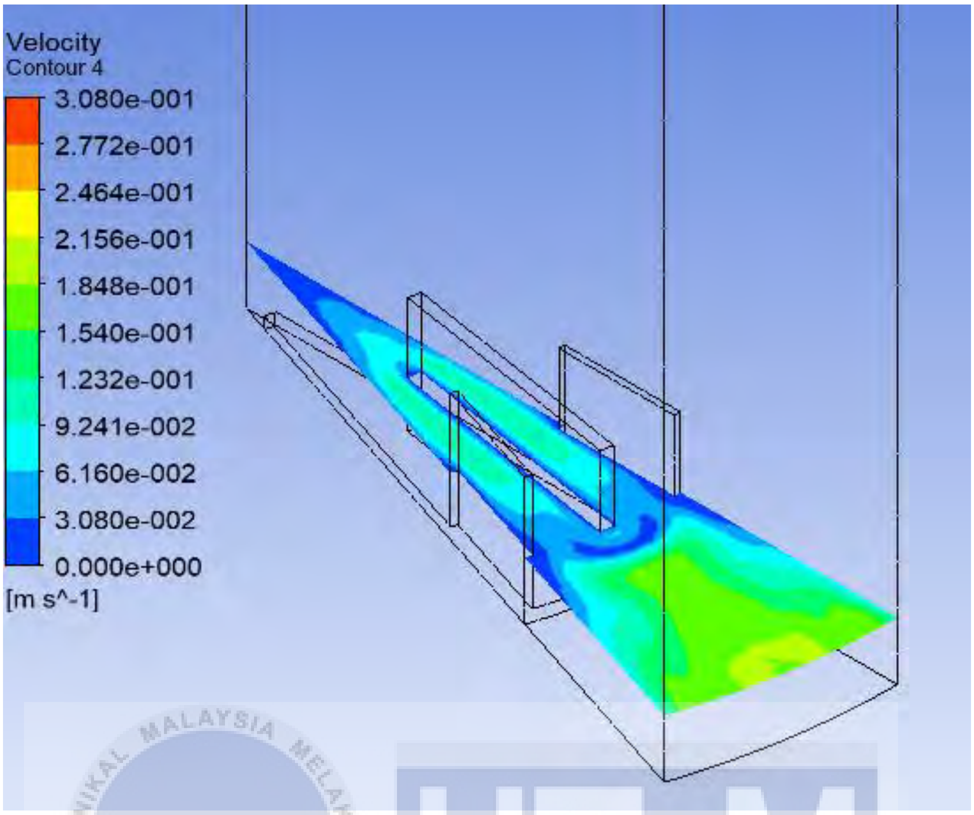


a) Length of fin 40/20 mm & 20mm height

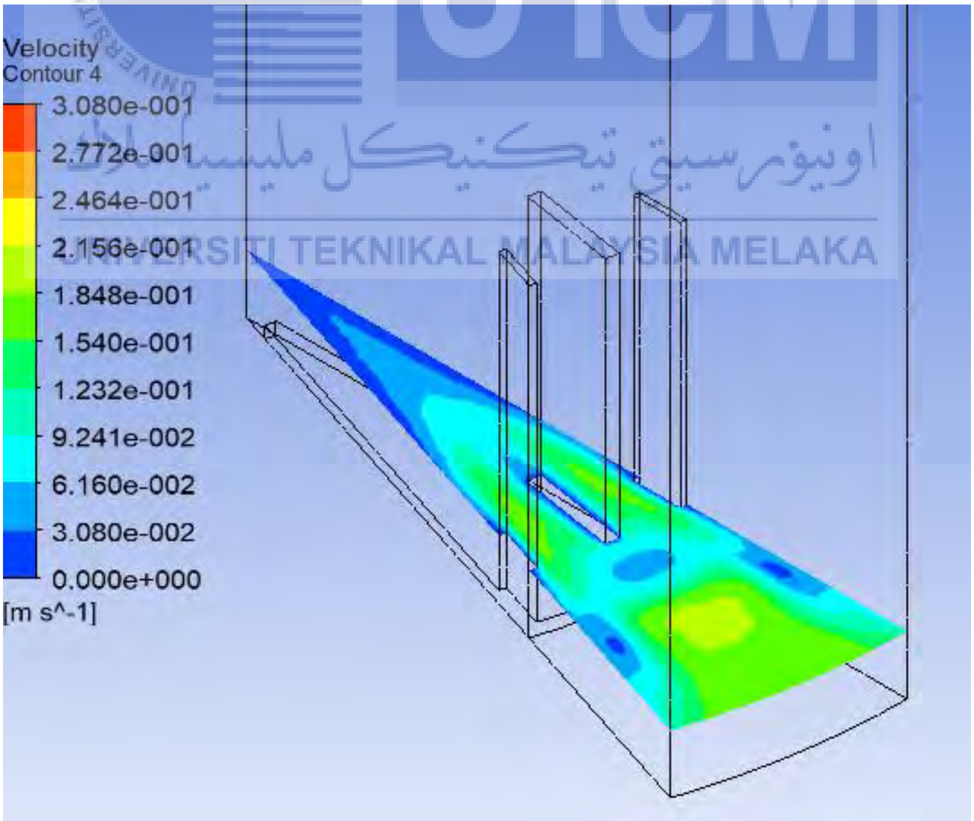


b) Length of fin 16/8 mm & 50mm height

Figure 4.6 : Temperature contour of heat sink at y = 10mm



a) Length of fin 40/20 mm & 20mm height



b) Length of fin 16/8 mm & 50mm height

Figure 4.7 : Velocity contour of heat sink at y = 10mm

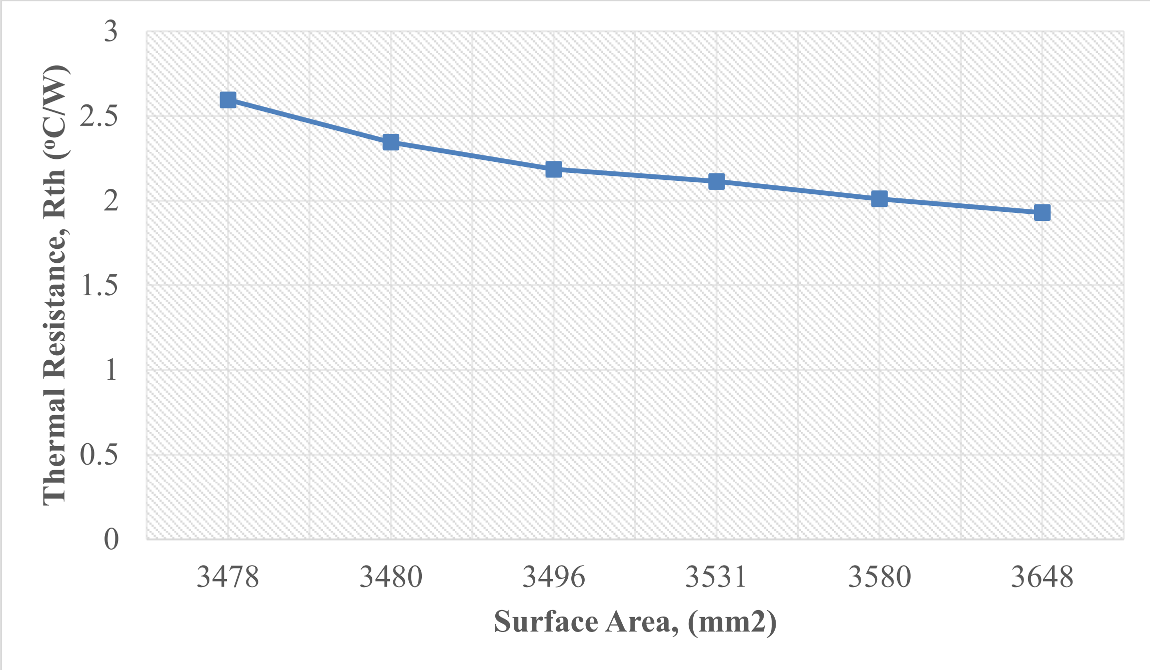


Figure 4.8 : Surface area of fin toward thermal resistance

4.5 Case Study 2 : Thermal Resistance Versus Thickness of Fin With Fixed Mass

In this section, the effect of fin thickness with fixed mass toward the thermal resistance will be discussed. Based on the result obtained from numerical analysis, when fin thickness increased, the thermal resistance of the heat sink increased. Increase in thermal resistance indicated that thermal performance of the heat sink decreased. Table 4.4 shows the result of the numerical analysis. The fin thickness shown in the table were the thickness of long fin.

Table 4.4 : Numerical analysis for case study 2

No	Thick -ness (mm)	Length of Height (mm)	Ratio of Thickness / Height	Mass ratio	Thermal Resistance , R_{th}	Increment percentage , %	Surface Area (mm^2)
1	1	40	0.0125	1.0000	1.6580	29.28	6620
2	2	20	0.0500	4159.4 mm^3 **reference**	2.3446	**reference**	3480
3	6	6.6667	0.3000	1.0003	5.0068	-113.54	1588

The result of numerical analysis for case study 2 was shown in figure 4.9. As can be seen from the figure, with fin thickness increased, the thermal resistance of the heat sink greatly increased. The minimum fin thickness of heat sink tested was 1mm, further decrease in the fin thickness will bring better cooling performance but mass production of the heat sink will be impossible according to Park et. al. (2016). The thickness of fin below 1mm was too small for the fabrication by die casting. While the maximum thickness of fin test was 6mm due to insufficient spaces for the radial heat sink.

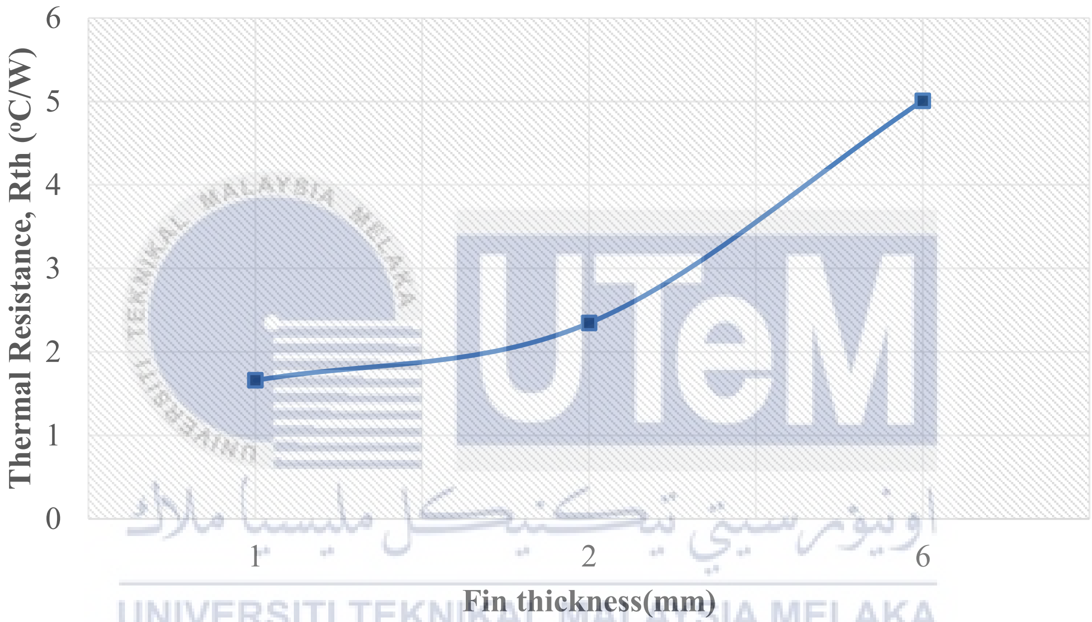
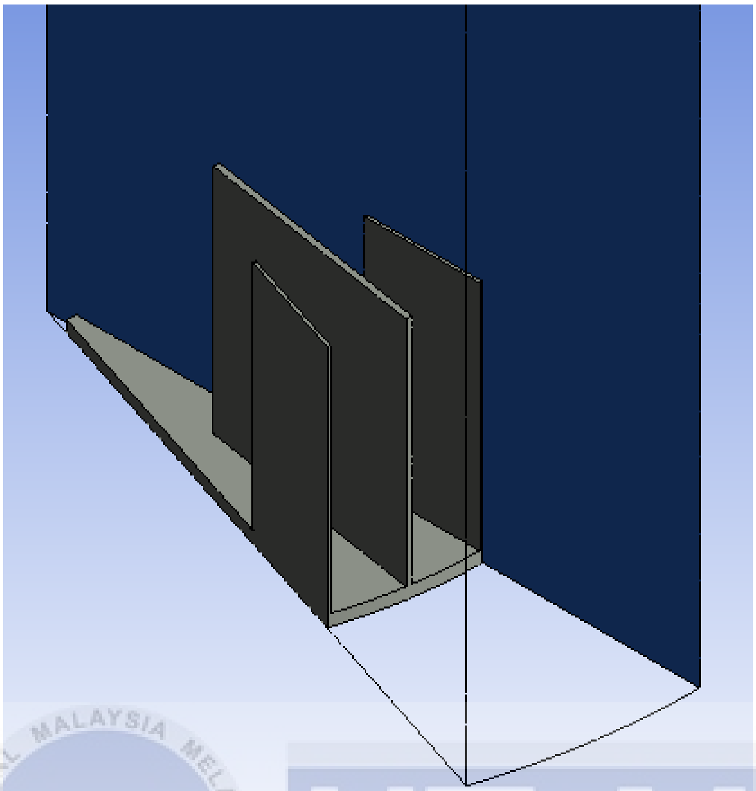
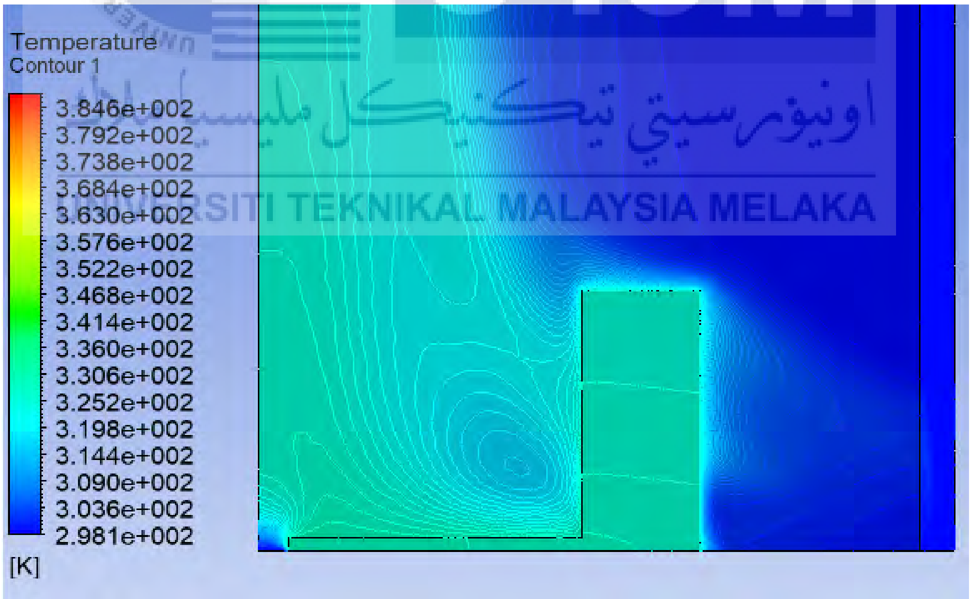


Figure 4.9 : Effect of thickness of fin toward thermal resistance

Based on table 4.4, fin thickness of heat sink increased, the thermal resistance of heat sink also increased. This indicated the cooling performance of heat sink decreased because of heat transfer surface area of the heat sink decreased when increasing in fin thickness. Increased in fin thickness will result in the height of fin reduced. With a surface area of fin greatly reduced, the heat transfer of heat sink became lower. This is shown in figure 4.10, the temperature of the heatsink was lowest for fin thickness of 1mm with a high height of fin while the highest temperature of the heatsink for the larger fin thickness 6mm.

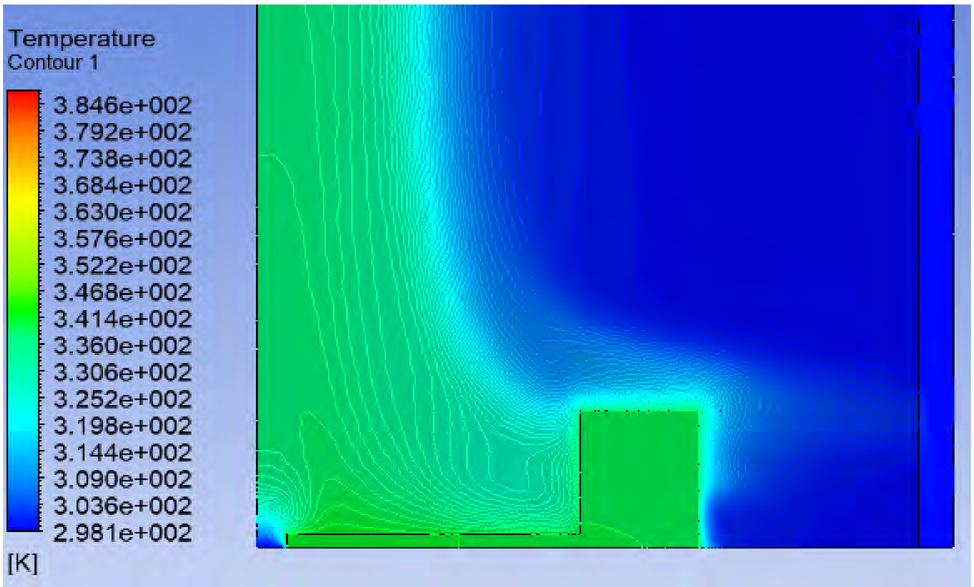


a) Plane of wall fluid for temperature contour

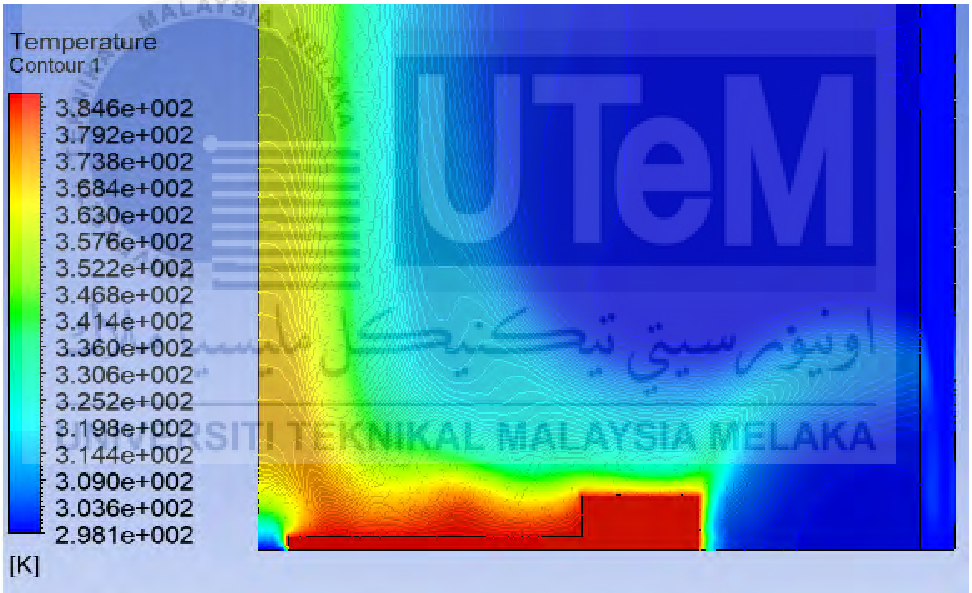


b) Fin thickness of 1mm

Figure 4.10: Temperature contour of the heat sink at wall fluid



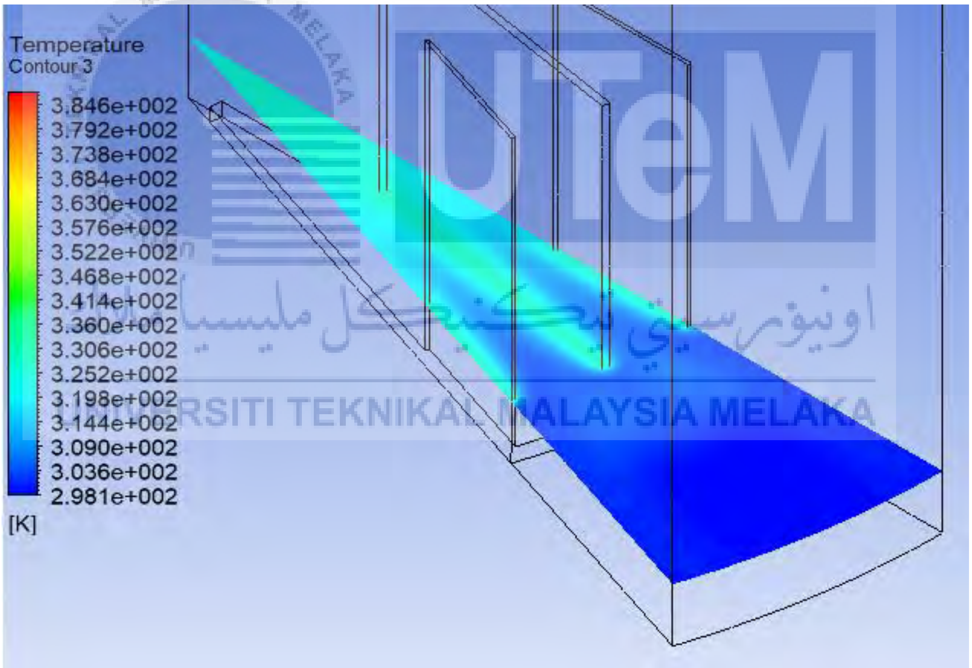
c) Fin thickness of 2mm



d) Fin thickness of 6mm

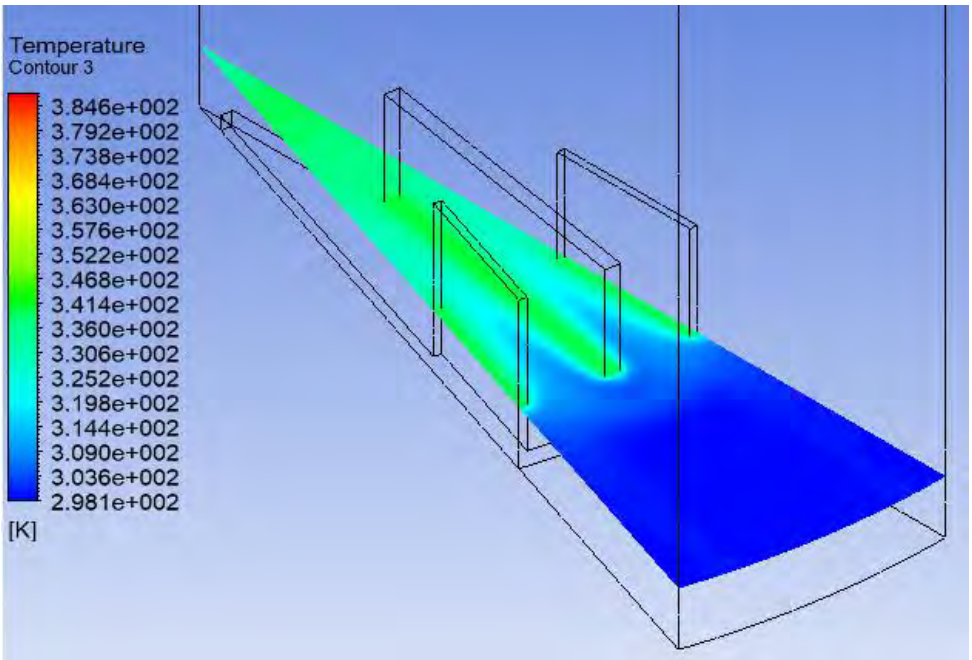
Figure 4.10: Temperature contour of the heat sink at wall fluid

Based on figure 4.11c, enlarged fin thickness caused the gap between other fins narrowed. With this, the thermal boundary layer tended to develop too early and overlapped with other fins which caused less airflow at outer region into the inner region of the heatsink. The inner region of heat sink remained at high temperature which causes heat transfer of heat sink became lower due to the low-temperature difference. With the fin thickness reduced greatly as shown in figure 4.11a, the inner region temperature of heat sink remained low. This is because thermal boundary layer between fins developed at a later stage which greatly increased heat dissipation when airflow from the outer region into inner region.

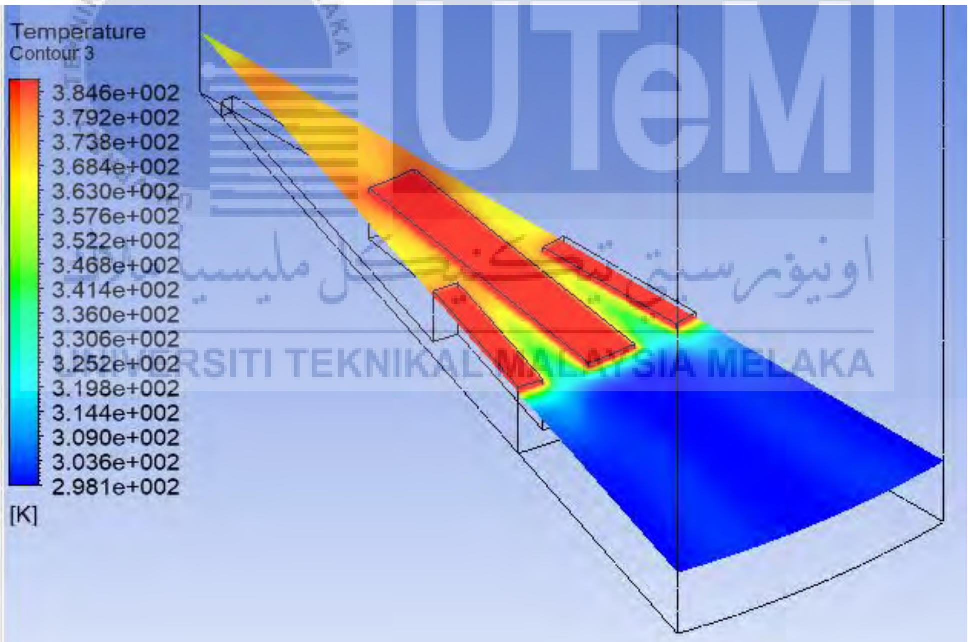


a) Fin thickness of 1mm

Figure 4.11 : Temperature contour of heat sink at y = 8 mm

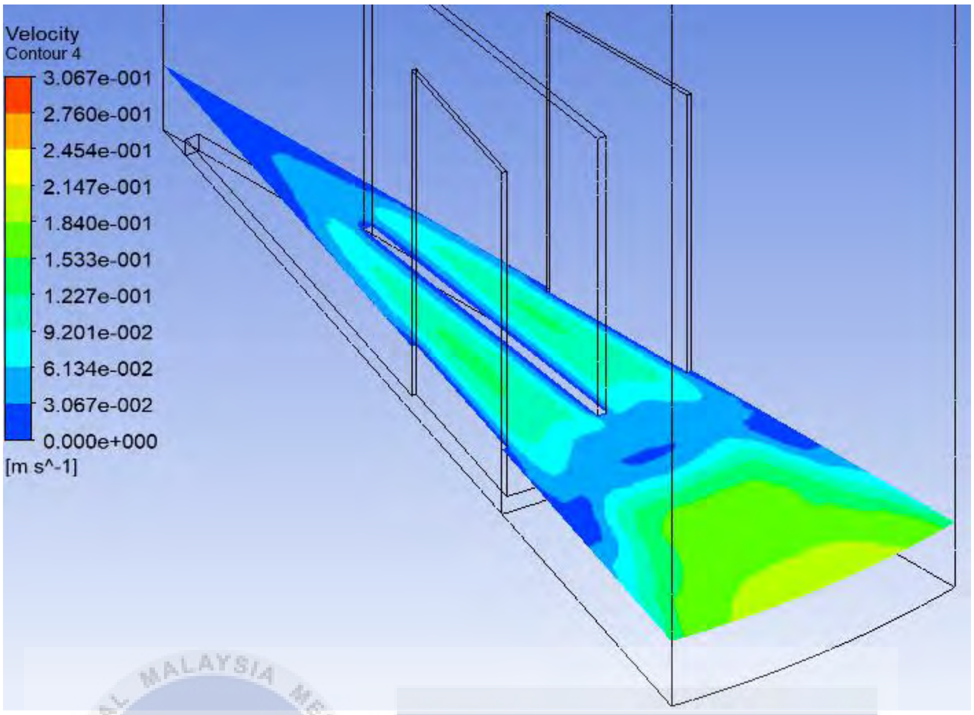


b) Fin thickness of 2mm

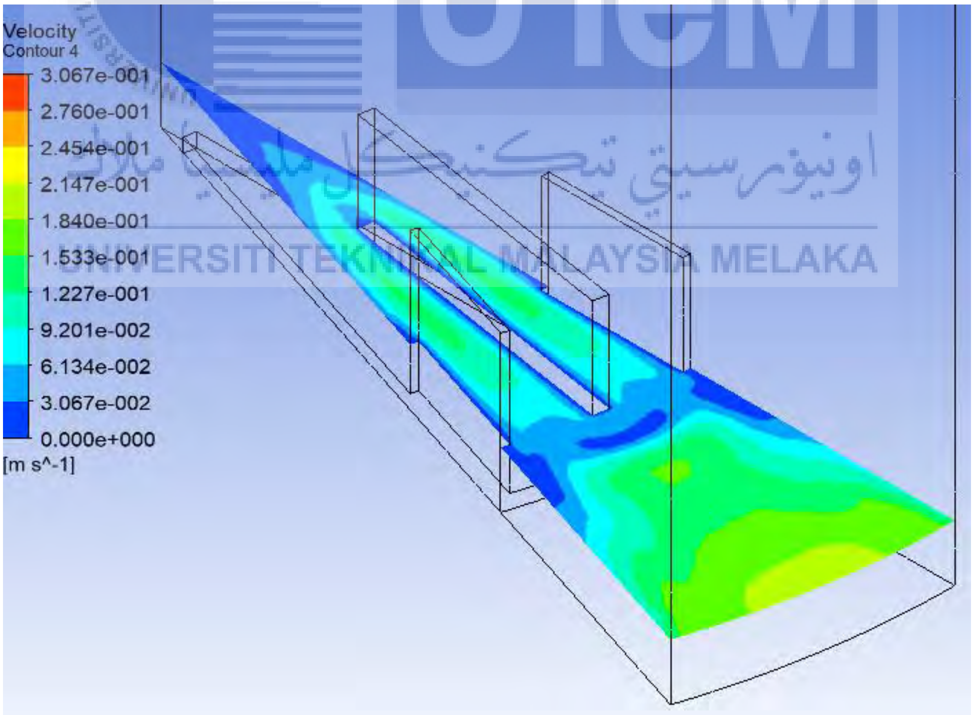


c) Fin thickness of 6mm

Figure 4.11 : Temperature contour of heat sink at y = 8 mm

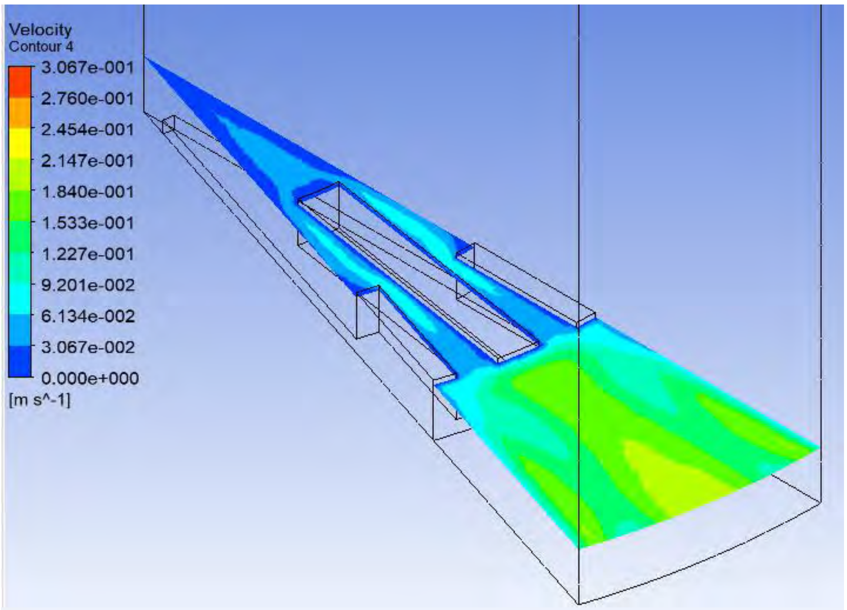


a) Fin thickness of 1mm



b) Fin thickness of 2mm

Figure 4.12 : Velocity contour of heat sink at y = 8 mm



c) Fin thickness of 6mm

Figure 4.12 : Velocity contour of heat sink at y = 8 mm

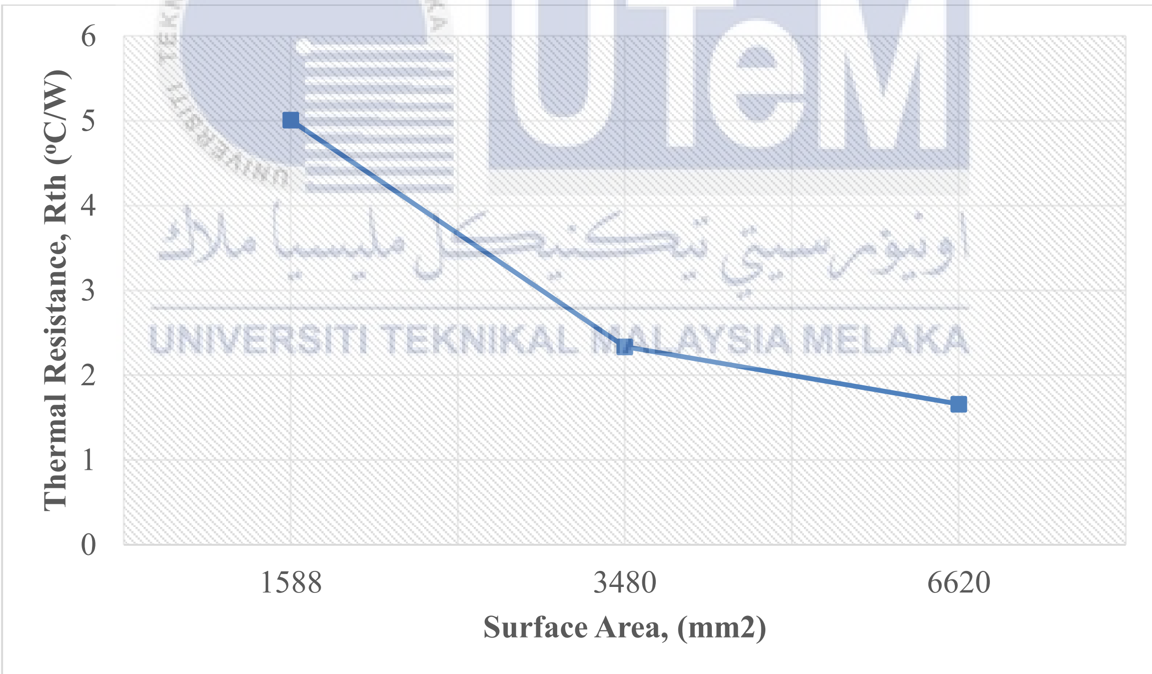


Figure 4.13 : Surface area of fin toward thermal resistance

Based on figure 4.13, with the decrease in thickness of fin, it increased the heat transfer surface area greatly. This is because increment in height of heatsink. Therefore, with increase of heat transfer surface area, the cooling performance of heatsink improved.

CHAPTER 5

CONCLUSION AND RECOMMENDATIONS

In this paper, geometry optimization of heatsink for LED lighting without increasing the cost of production was done in the study. Without increasing the cost of production mainly indicated that there was no excess material used when mass production of the LED heat sink. In order to do that, studies of geometric parameter of heat sink were conducted. By using numerical analysis method, Ansys software was used to simulate the result of geometry changes.

Throughout the whole studies, the cooling performance of heat sink was highest efficient when the thickness of fin was small while the height of fin was high. With the condition met, the heat sinks able to transfer heat efficiently without waste of excessive material. This is because when the thickness of fin reduced, the thermal boundary layer of heat sink developed almost at the inner region of heatsink. Which meant more outer region air able to flow into the inner region to maintain inner region of heat sink temperature remained low. With huge temperature difference between heat sink and air at inner region, heat transfer tended to transfer better than others. With the height of fin increased, the surface area of fin greatly increased. With the increment in surface area, heat transfer surface area also greatly increased which led to cooling performance improved significantly.

Lastly, I would like to recommend that this study not only applied to radial heat sink but also a rectangular heat sink. Since rectangular heat sink was widely used in many aspects such as cooling of internal components of the computer. Geometry parameter to study would be recommended to be fin thickness and height of fin which it greatly improved the thermal performance of heat sink compared to another geometry parameter like the length of heatsink.

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