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Original Research Article

Investigation and Optimization of Heat Removal from a Micro-Processor Using Solid Works 2013 and Ansysy Workbench

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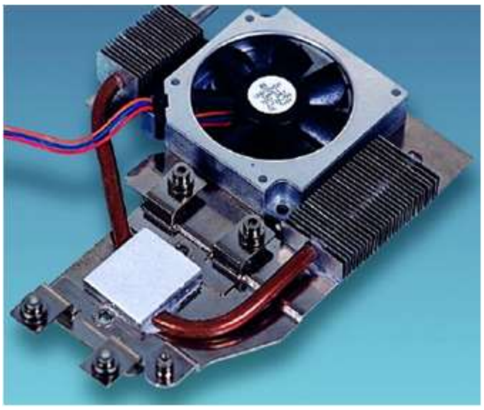
Abstract: The global warming increase especially in the tropical regions like west Africa countries like Nigeria calls fall optimization of heat removal system to avoid regular failure of electronics for repairs or permanent damage. Microprocessors work best at low temperatures, the need to condition its ambient condition at low temperatures at minimum of (29 -38)0c is necessary. The effective efficiency and performance of an electronic Circuit is an electronic appliance depends on the condition of its environment. Most electronic components do not function best under high temperatures as their performance is inversely proportional to temperature increase. The use of external fans to cool electronics when in use is energy and cost sapping. Hence, this paper tends to design and optimize heat removal rate from a microprocessor using heat tube and cooling fins incorporated with a high power suction fan of 0.75watt with 40x40x1.5 fins, increase in surface area, number of fins and suction speed of the fan are parametric factors that improve rate of heat removal. Designed and developed with solidworks 2013 and Ansys Work Benchmultiphysics, results show a directional heat flux $2.8117 \times 10^6\ W/m^2$ and a total heat flux of $5.7828 \times 10^6\ W/m^6$ in the system.
Keywords: Heat Removal rate optimization; Heat Tube, Cooling Fins, Suction Fan and Micro Processor

INTRODUCTION

Heat transfer is the area which describes the energy transport between material bodies due to a difference in temperature, and its development and applications are of fundamental importance in many branches of engineering since provides economical and efficient solutions for critical problems encountered in many engineering items of equipment.. Electronic components are class of semi-conductors -diodes, transistors, IC and micro-processor which functions best at low temperatures. The manufacturers of these products though may aware of the effect of heat on their performances but they are manufactured based on their prevailing climatic conditions. For reliability and durability is the use of these appliances in the tropical region like Africa coupled with the increase in world temperature resulted from the depletion of ozone layer. It is necessary to design and optimize a heat removal medium that will facilitate favorable temperature reduction. The world is fond of electronic appliances, the steady and continuous use can only be guaranteed if there is an optimized and dependable in-build heat removal.

Heat energy flows due to temperature gradient. Hence, conduction, convection and radiation are processes through which heat is transferred. However, thermal management in the portable electronics environment is becoming increasingly difficult due tohigh heat load and dimensional constraints. Proper selection of fans and fin pitch in the heat sink is crucial to ensure the thermal design of the system is optimized Heat energy relevance in power generation is steam and gas turbine, transportation and in industrialization cannot be over emphasized; but its desirable effects on electronic products, surfaces of products, material, surfaces of products, and material composition of products provoke an engineering ingenuity to probe into, action, protecting and curbing. These measures are taken individually or combined to salvage the life span of an engineering material and its performance (functionality).



a.

b.

Fig-1.0: Blower-heat sink system for an electronic appliance.

Figures 1.0 show thermal solutions found in today's portable computer and flat screen TV environment. The mini plastic blower-heat sink heat pipe assembly shown in Figure 1.0 is mounted on a slim 4mm circuit cold plate. Heat pipes made of copper transport the heat generated by the chip or micro-processor, where it is dissipated to the environment with the aid of the airflow produced by the mini blower. This type of thermal design is deployed in a system with limited space. The fin designs are optimized to assure maximum heat transfer occurs between the fins and the surrounding ambient. The mini blower is capable of supplying sufficient airflow to cool the system while maintaining a low profile to fulfill the space restriction [1]. The axial fan- heat pipe-heat sink assembly shown in Figure 1.0 is used when space is less constrictive. A heat pipe transports heat from the source to the high efficiency fins enclosed in a square block with an axial fan attached at one end to pull airflow through the fins. One of the objectives of this paper is to investigate the effects on thermal performance under axial fan and micro blower airflow for different gap/length ratio heat sinks. In doing so, the optimization point between the different types of fans or blowers under certain fin gap/length ratio can be obtained.

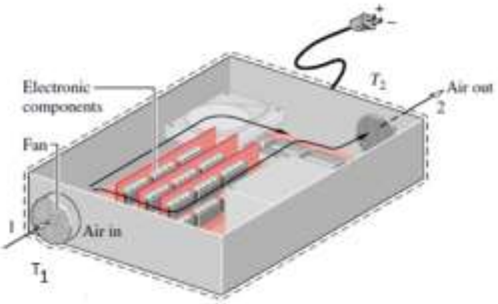


Fig.2 cooling of electronic components

The design under analysis is more robust and compacted than the analogy in fig.2.0 because it is two suction fans and may increase production cost.

Scope

The work embraces cad and CAE work using knowledge base software solid works and Finite Element Analysis to evaluate and optimize rate of heat removal from a micro-processor and that parametric factors that will facilitate this process in a heat tube and cooling fins with suction fan assembly.

Aim and Objective

Temperature reduction on electronic components like microprocessor is vital for its functionality and performance hence his research work conducted with the following aims and objectives

1. To increase heat removal rate from the micro-processor
2. To curb or prevent failure of the component
3. To encourage the use of electronics through a well-designed and optimized heat removal medium.
4. To guarantee steady and long use of electronic products (durability and longitivity)

Contribution

Using knowledge base software Solid Works and tropical climatic temperature parameters as boundary conditions to optimize an existing heat removal system that will guarantee the effective used, reliability and durability of electronics in African regions characterized with high temperatures besides the increase in the world temperature from depleted ozone layer. Solutions and suggestions will be made.

Significance

The essence of this work is to achieve good temperature reduction rate within the ambient of microprocessor to guarantee its performance and efficiency as well as its durability and longentivity.

Heat Energy Flow in the Fluid Stream

The plastic fan hydraulic diameter and channel velocity necessary to draw heat from the micro processor via the tube (ducts) to the cooling fin are determined as

$$H_d = \frac{2g\,L_f}{g + L_f} \qquad (1)$$

Where

$H_d = hydraulic\ diameter\ (m)$

$g = channel\ width, (m)$

$L_f = hieght\ of\ the\ fin,$(m)

$$V_g = V_{fs}\left(1 + \frac{t}{g}\right) \qquad (2)$$

$\mathrm{Vg} = \text{ Channel velocity}, \frac{\mathrm{m}}{\mathrm{s}}, \mathrm{Vfs} = \text{Free stream velocity}, \mathrm{m/s}$

$\mathrm{t} = \text{Fin thickness}, \mathrm{m}$

V_g is a function of free stream velocity and ratio of fin thickness and channel width. The fluid variables of conducting tube has a Reynolds number of

$R_{eg} < 2300$

$$Re_g = \frac{H_d V_g}{v} \qquad (3)$$

$\mathrm{Reg} = \text{ Channel Reynolds number}$

$\mathrm{v} = \text{Fluid viscosity}, \mathrm{m2/s}$

in which Since the Reynolds number is below 2300, the velocity profile in the heat sink channel is laminar, and in case of these compacted appliances like computers (lap tops) and flat screen TV of LCD or LED types, the length of the tube is not long enough to have fully developed laminar flow hence a mixture of fully developed and developing flow occur. The work of Shah and London [2] shows that the apparent friction factor for such a mixture of flow is a function of friction factor of the fully developed flow and the hydrodynamic entrance length. The approximate equation is

$$f_{app}Re_g = \left[\left(\frac{3.2}{(X^{+})^{0.57}}\right)^2 + \left(fRe_g\right)^2\right]^{1/2} \qquad (4)$$

$f_{app} = \text{Apparent friction factor}$

X^+ = Dimensionless hydrodynamic entry length
fRe_g = fully developed flow friction factor
f Reg, is a fully developed flow friction factor obtained from [3]. Since the flow is laminar, the dimensionless hydrodynamic entrance length is defined as a function of Reynolds number, hydraulic diameter and the lengthof the channel as follows [4]:

$$X^+ = \left(\frac{X_1}{Re_{H_d} H_d}\right)_{laminar} \quad (5)$$

X_1 = Channel length, m
Re_{H_d} = Renold number for hydraulic diameter
The laminar flow contraction and expansion loss coefficients
are defined as [5]:
$K_c = 0.8 - 0.4(g/w)^2$ (6)
$K_e = [1-(g/w)]^2 - 0.4(g/w)$ (7)
Kc = Contraction coefficient
Ke = Expansion coefficient
w = Fin pitch, m

SYSTEM HEAT PRESSURE DROP

Several factors affect the performance of heat removal system. These constraints are hydrodynamic head, geometry of heat sink and fluid properties imposed upon the system [6]. When the fan performance is balanced by the system performance to deliver the most efficient combination of airflow and heat sink surface area. The accelerating flows inside the heat removal system produce a pressure drop between the channel entrance and exit region. This pressure drop across the heat sink is also known as the system resistance. System resistance affects the overall thermal performance of heat sink. Higher system resistance causes less airflow through the heat sink channel, attaining lower convection heat transfer rate between the fins and the surrounding air and increases the fin thermal resistance. When the heat sink system resistance is known, the actual volumetric flow rate can be found from the fan/blower performance curve with a given total heat sink pressure drop. The total heat sink pressure drop is formulated as:

$$\Delta P = \left(K_c + 4f_{app}X^+ + K_e\right).\left(\frac{\rho {V_g}^2}{2}\right) \quad (8)$$

ΔP = Heat sink pressure drop, N/m^2
f_{app} = Apparent friction factor

Vg = Channel velocity, m/s
ρ = Fluid density, Kg/m^3
The actual approach velocity can be calculated by dividing the volumetric flow rate
with duct cross sectional area.
$V_a = Volumetric\ Flow\ Rate / A_d$
Va = Actual approach velocity, m/s
A_d = Duct cross sectional area, mm2

RESISTANCE TO THE FLOW OF HEAT

The total thermal resistance is the product of the fin thermal resistance and the base spreading resistance, expressed as follow:
$\theta_T = \theta_f + \theta_s$ (9)
θ_T = total thermal resistance
θ_f = fin thermal resistance
θ_s = base spreading thermal resistance

However the resistance offered to the heat transported and transferred through the conducting tube (duct) should not be neglected, hence
$\theta_T = \theta_f + \theta_s + \theta_d$ (10)
θ_d = conducting tube(duct) thermal resistance

Fin Thermal Resistance

With the adiabatic fin tip assumption, the fin thermal resistance is given by:

$$\theta_f = \frac{1}{\eta_f A_f h} \qquad (11)$$

h = Convection heat transfer coefficient, W/m2K

A_f = Fin surface area, mm2

$\eta_f = efficiency\ of\ the\ fin$

the coefficient of heat transfer, h, is determined by the application of approach velocity Reynolds numberaRe ,The approach velocity Reynolds number is evaluated by taking the aspect ratio of the channel width to length and it is defined as:

$$Re_a{}^* = \frac{V_a g}{\nu}\left(\frac{g}{L}\right) \qquad (12)$$

$Re_a{}^*$ = Approach velocity Reynolds number

Va = Actual approach velocity, m/s

L = Fin length, m

ν = Fluid viscosity, m2/s

g = Channel width, m

Since the flow in the system duct is a mixture of fully developed and developing laminar, the composite model proposed by Teertsra is used to calculate the average Nusselt number in the channel [7]:

$$Nu_g = \left[\left(\frac{Re_a{}^* Pr}{2}\right)^{-3} + \left(\sqrt{Re_a{}^* Pr^{\frac{1}{3}}}\sqrt{1+\frac{3.65}{\sqrt{Re_a{}^*}}}\right)^{-3}\right]^{\frac{1}{3}} \qquad (13)$$

Nug = Nusselt number

Pr = Prandtl Number

the coefficient of heat transfer, h, can be expressed as:

$$h = \frac{Nu_g K_{air}}{L} \qquad (14)$$

kair = Air thermal conductivity, W/Mk

the fin width is sufficiently large compared with the fin thickness, and along with the coefficient of heat transfer and fin geometry, the fin efficiency, ηf, is given as [8]:

$$\eta_f = \frac{tanh mL_c}{m L_c} \qquad (15)$$

wheremLc is defined as:

$$mL_c = \sqrt{\frac{2h}{K_m A_m} L_c{}^{2/3}} \qquad (16)$$

km = Material thermal conductivity, W/mK

Lc = Corrected length, m

Am = Fin profile area, m2

and the corrected fin length and fin profile area are found using these equations

$$L_c = L + \frac{t}{2} \qquad (17)$$

$$A_m = L_c t \qquad (18)$$

Base Spreading Resistance

As heat flows across the cross sectional area of the base, it encounters resistance and therefore gives rise to the base temperature. Depending on the size of the heat source, the smaller the heat source, the higher the base spreading resistance. The empirical solution by Lee is shown as follows [9]:

$$\theta_s = \left(\frac{\sqrt{A_b}-\sqrt{A_{sc}}}{K_m\sqrt{\pi A_m A_{sc}}}\right).\left(\frac{K_m A_b \theta_{ave} + tanh(\lambda t_b)}{1+\lambda k_m A_b \theta_{ave} tanh(\lambda t_b)}\right) \qquad (19)$$

Ad = Duct cross sectional area, mm2

$\mathrm{Ab} = $ Heat sink base area, mm2
$\lambda = $ Spreading resistance variable, m − 1
$\mathrm{Af} = $ Fin surface area, mm2
$\mathrm{Am} = $ Fin profile area, mm2
$\mathrm{Asc} = $ Heat source contact area, mm2
$\Theta\mathrm{ave} = $ Average heat sink thermal resistance, °C/W
$\mathrm{Tamb} = $ Ambient temperature, °C
$\mathrm{Tb} = $ Heat sink base temperature, °C
$\Theta\mathrm{f} = $ Fin thermal resistance, °C/W
$\Theta\mathrm{s} = $ Base spreading resistance, °C/W
$\Theta\mathrm{t} = $ Total resistance, $° \frac{\mathrm{C}}{\mathrm{W}}$
where λ is given as:

$$\lambda = \frac{\pi^{\frac{3}{2}}}{\sqrt{A_b}} + \frac{1}{\sqrt{A_{sc}}} \qquad (20)$$

the average thermal resistance, Θave, is assumed to be equal to the fin thermal resistance, Θf.

FIN ANALYSIS

The magnitude of the temperature gradient decreases with increasing *x*. This trend is a consequence of the reduction in the conduction heat transfer $qx(x)$ with increasing *x* due to continuous convection heat transfer from the surface. The *fin heat rate* may be evaluated in two alternative ways, both of which involve use of the temperature distribution. The first involves applying *Fourier's law* at the fin base, *x* _ 0,

$$Q = KA\left(\frac{dT}{dx}\right)_{x=0} \qquad (21)$$

Fin Performance Parameters

Since fins are used to increase the heat transfer rate from a surface by increasing the effective surface area. The fin itself represents a conduction resistance to heat transfer from the original surface. For this reason, there is no assurance that the heat transfer rate will be increased through the use of fins. An assessment of this matter may be made by evaluating the ***fin effectiveness***$\boldsymbol{\varepsilon_f}$, which is defined as the *ratio of the fin heat transfer rate to the heat transfer rate that would exist without the fin*

$$\varepsilon_f = \frac{q_f}{hA_c\theta_b} \qquad (22)$$

where ,*Ac* is the fin cross-sectional area.
Fin performance may also be quantified in terms of a thermal resistance. Treating the difference between the base and fluid temperatures as the driving potential, a ***fin resistance*** may be defined as

$$\mathrm{R_{t,f}} = \frac{\theta_\mathrm{b}}{\mathrm{q_f}} \qquad (23)$$

This result is extremely useful, particularly when representing a finned surface as a thermal circuit element. Note that, according to the fin tip condition, the appropriate expression for qf is obtained from Dividing Eq. 23 into the expression for the *thermal resistance* due to convection at the exposed base,

$$R_{t,b} = \frac{1}{hA_c} \qquad (24)$$

and substituting from Eq. 22 it follows that

$$\varepsilon_f = \frac{R_{t,b}}{R_{t,f}} \qquad (25)$$

The fin effectiveness may be interpreted as a ratio of thermal resistances, and to increase εf it is necessary to reduce the conduction–convection resistance of the fin. If the fin is to enhance heat transfer, its resistance must not exceed that of the exposed base. Another measure of fin thermal performance is provided by the *fin efficiency* η*f*. The maximum driving potential for convection is the temperature difference between the base (*x* _ 0) and the fluid

$$\theta_b = T_b + T_\infty \qquad (26)$$

Hence the maximum fin heat rate is the rate that would exist *if* the entire fin surface were at the base temperature:

$$q_{max} = hA_f\theta_b \qquad (27)$$

Where Af is the total surface area of the fin. However, since any fin is characterized by a finite conduction resistance, a temperature gradient must exist along the fin and the above condition is an idealization. A logical definition of ***fin efficiency*** is therefore

$$\eta_f = \frac{q_f}{q_{max}} = \frac{q_f}{h A_f \theta_b} \qquad (28)$$

DESIGN ANALYSIS

Methodology

The heat flux analysis is conducted using solid works with the boundary conditions of temperature ranging from 34^{o}C to 80^{o}C and optimization conducted with the following material selection

- Plate adapted on the processors pure aluminium 2.5mm thick
- Heat tube-pure copper with 0.8mm thick with diameter 8mm
- Cooling fins- pure aluminium 1mm thick; a cross sectional area of 30×30mm

The fan is a radial suction fan made up of plastics besides the BLDC motor. Measurements were taken from LCD and LED panels in electronics repairing firm in the City of Enugu in Nigeria

Assumptions

1. Steady state uniform heat transfer
2. The duct cross sectional area is uniform
3. The base heat is the heat from the source –micro processor heat generated.
4. The suction blower is at constant speed of 1200rpm

Heat Generated By the Micro Processor

The heat generated is assumed to be Q with temperature reference of the prevailing temperature as the maximum temperature of the hot season as T_0 as norma temperature of $34^0\,C$ the is due to the rise in world's temperature.

Models

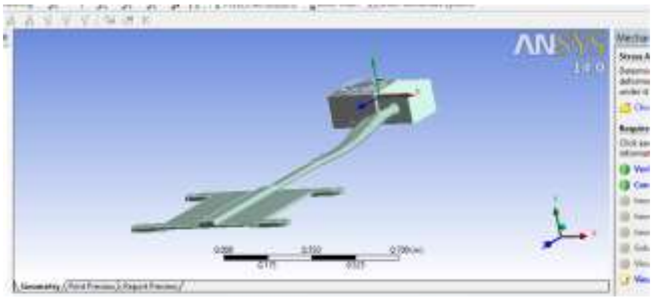


Fig-3: 3-D model of heat sink system

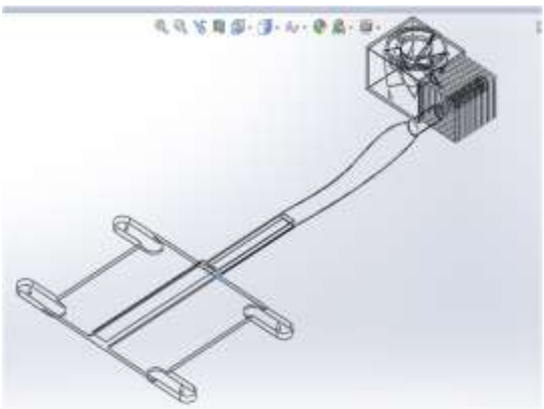


Fig-4: 3-D WIRE FRAME VIEW

The finite element analysis FEA for heat conduction from the plate, tube and fins and convection by the fans is modeled to x-ray the processes.

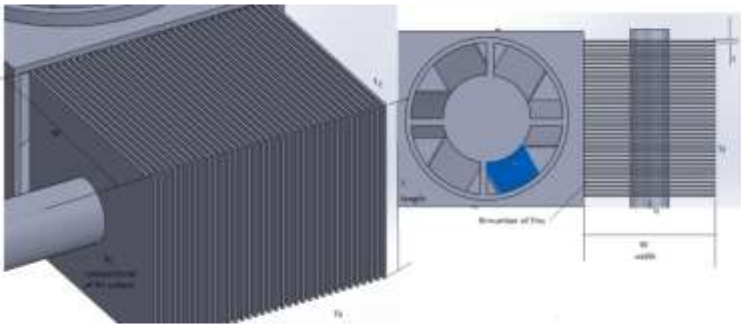


Fig 5.descriptive model for the mathematical demonstration

The total heat flused is the heat from the unfinned surface plus the heat flux from the fins

$$Q = Q_u + Q_f \qquad (29)$$

$$Q_U = A_S h\left(T_b - T_f\right) = w \times s(N-1)h\left(T_b - T_f\right) \qquad (30)$$

For a single fin the surface temperature T

$$T - T_F = \frac{(T_b - T_f)coshm(L-x)}{sinhmL} \qquad (31)$$

$$\frac{dT}{dx} = \frac{-sinhm(L-x)}{coshmL} m\left(T_f - T_f\right) \qquad (32)$$

$$Q_f = -KA_c\left(\frac{dT}{dx}\right)_{x=0} \qquad (33)$$

$$Q_f = KA_C m\left(T_b - T_f\right)tanh(ml) = (hpkA_c)^{1/2}\left(T_b - T_f\right)tanh(mL) \qquad (34)$$

$$m = \left(\frac{hP}{KA_c}\right)^{1/2} \qquad (35)$$

$$P = 2(w + t) \qquad (36)$$

From

$$Q = Q_u + Q_f \qquad (37)$$

The total heat flux becomes

$$Q = \left(A_S h\left(T_b - T_f\right) = w \times s(N-1)h\left(T_b - T_f\right) + \left(KA_C m\left(T_b - T_f\right)tanh(ml) = (hpkA_c)^{1/2}\left(T_b - T_f\right)tanh(mL)\right)\right) \qquad (38)$$

Fin effectiveness

$$\varepsilon_{fin} = \frac{fin\ heat\ transfer\ rate}{heat\ transfer\ rate\ that\ would\ occur\ in\ absence\ of\ the\ fin} \qquad (39)$$

Fin efficiency

$$\eta_{fin} = \frac{actual\ heat\ energy\ transfer\ through\ the\ fin}{heat\ that\ would\ have\ been\ transfered\ if\ the\ fin\ area\ were\ at\ the\ base\ temperature} \qquad (40)$$

$$\eta_{fin} = \frac{Q_f}{hA_s(T_b - T_f)} \qquad (41)$$

$$A_s = wL + wL + Lt + Lt = 2L(w + t) \qquad (42)$$

MODELS

Considering a single fine model, the simulation is run with boundary conditions using solid works

T_b=60^{0}C
T_b=20^{0}C

Table-1: Thermal Properties Aluminium

Density	7850 kg m^-3
Coefficient of Thermal Expansion	1.2e-005 C^-1
Specific Heat	434 J kg^-1 C^-1
Thermal Conductivity	2700 W m^-1 C^-1
Resistivity	1.7e-007 ohm m

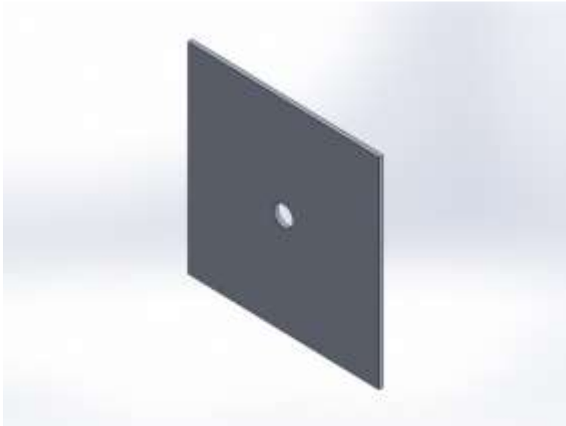


Fig-6: 3-D Model of A Fin Model

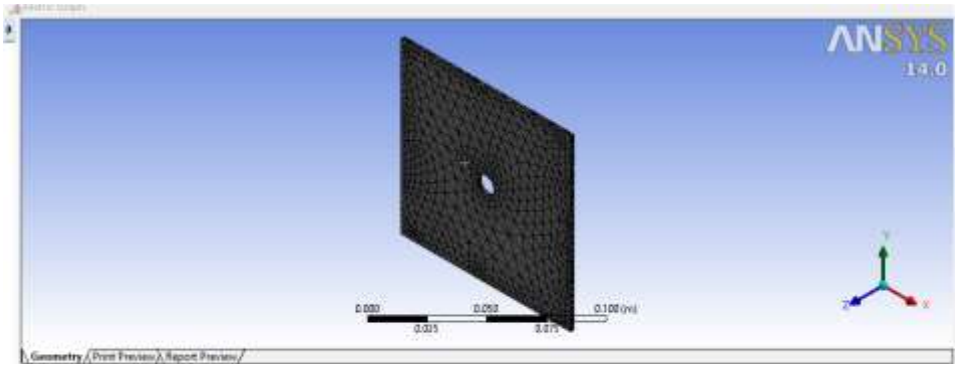


Fig-7: 3-D Model Of meshed Fin Model

Table-2: MESH PROPERTIES

Total Nodes	Aspect Ratio	Jacobian Points
15554	5.3811	4 Points
Total Elements	Mesh Type	Element Size
7788	Solid Mesh	2.8779 mm

SIMULATED MODELS

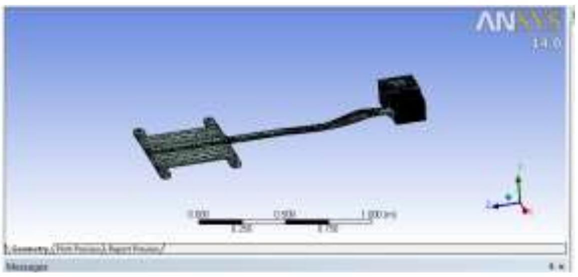


Fig-9: Solid mesh of the system model

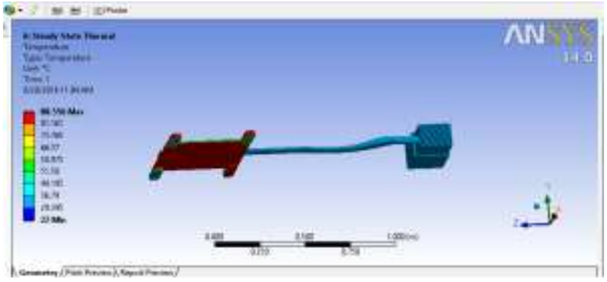


Fig-8: Temperature distribution in the heat sink system

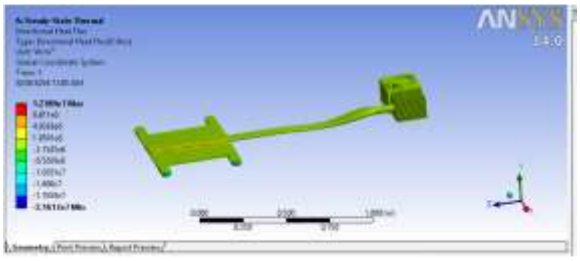


Fig-9: Directional heat flux distribution in the heat sink system

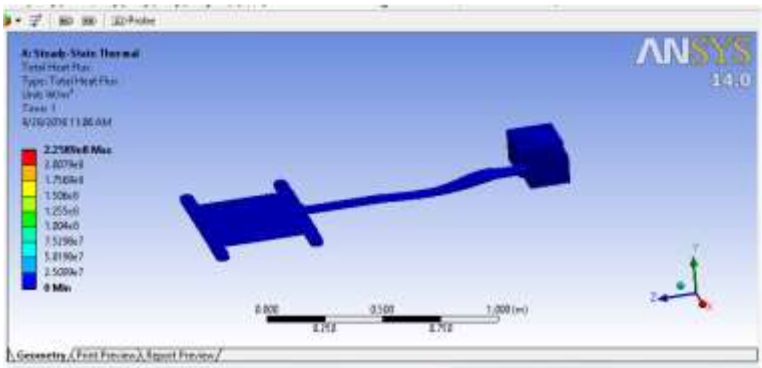


Fig-10: Total heat flux distribution in the heat sink system

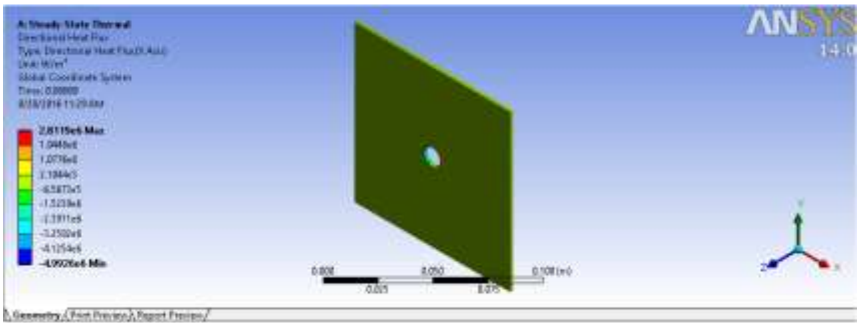


Fig-11: Directional heat flux distribution in a fin

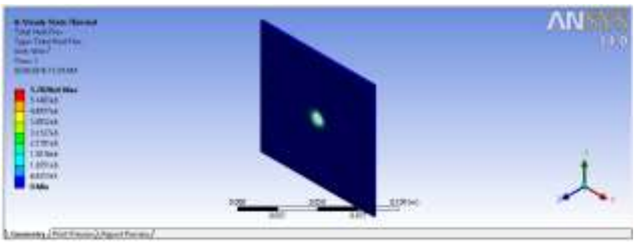


Fig-12: Total heat flux distribution in a fin

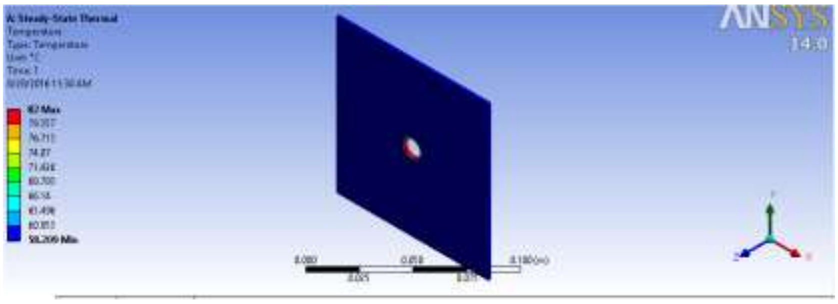


Fig-13: Temperature distribution in a fin

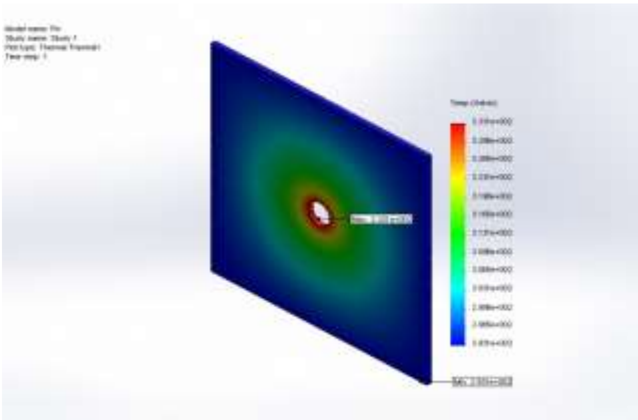


Fig-14: Simulated model of the fin with solid works 2013

DISCUSSION

The rate of heat transfer is dependent on factors such as the material type, like this heat sink method, the rate of transfer as governed by the suction fan, the cross-sectional area of the fins, the numbers of fins, the thickness of the fine and the material of the heat tube. Pure aluminum should be used for its thermal conductivity, absorption and diffusivity. Also for severe temperatures being experienced these days a constant speed fan may not be very helpful and a speed controller interface should be design for a variable speed motor programmable with temperature sensors to increase the speed as temperature defines.

CONCLUSION

An equitable heat sink system needs to be designed and optimized for electronics in hot regions of the world. The heat transfer rate increases with increase in crossectional area of the fin and decrease with increase in fin thickness, the speed of the fan will also increase the transfer rate. The material type is not undermined. The heat sink parametric factors need to be optimized to ensure the survival of electronics in this hot region of the world.

Acknowledgment

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