Computational Heat Transfer Analysis in Electronics using Piezo electric Fan Arrays

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Abstract*: Thermal design of miniaturized electronic equipment is becoming challenging with increased heat fluxes, acoustic disadvantages with enhanced noise from fans that cause forced convective cooling and also restriction in space for the placement of fans itself. Present study aims at evaluating the efficiency of a micro piezo electric fan in cooling a heat sink (20*20) mm^2 provided with an array of pin fins (6*6). FLOTHERM is used to do the CFD analysis and heat transfer analysis to determine the optimized spacing of fins.It is found that the best configuration has reduced the hot spot temperature from 167^{0}C to 64.1^{0}C.

Index Terms: FLOTHERM, Piezo electric fan, Thermal cooling in electronics

Nomenclature

n_f	Number of fins
C_f	Centre gap spacing on heat sink
u,v & w	Velocity components in x, y & z directions
P	Pressure
T	Temperature of solid/fluid materials
	Density
C_p	Specific heat
L	Length of heat sink
t	Thickness of fin
w	Width of fin
λ	Thermal Expansivity
Γ	Thermal diffusivity
φ	Thermal Dissipation function

I. INTRODUCTION

Electric rotational fans are utilized as convection cooling gadgets in electrical components, such as CPUs, power boards etc. But in modern electronic industry, electric component size is reducing and heat generation rate is increasing. But reducing rotary fan into such a small size is very difficult due to its complex structure and it is difficult to make a 10mm electric rotational fan. For such cases, an alternative to rotational fan is piezo electric fan, because of its simple parts, and it also satisfies the requirements of light weight, low noise and low power utilization.

A piezo electric fan comprises of piezo electric material attached to a flexible cantilever beam. An alternating input signal makes the piezo electric material to contract and expand, producing shear powers among itself and cantilever beam is produces motion at the free end of cantilever, which enhances heat exchange at that point when contrasted with natural convection heat transfer alone.

Numerous ways of insertion of piezo electric fans in cooling circuits have been examined in the literature. Kimber et al[1] conducted heat transfer experiments for two separate orientations of piezo fan (i.e., horizontal and vertical) and observed coupling phenomenon for two piezo fans oscillating in close proximity. R.R. Schmidt [2] recognized the best cooling area of piezo fan by deciding mass exchange coefficients tentatively.Tolga Acikalin et al,[3] conducted flow visualization experiments to understand the material science of fan operation. They used mobile phone and laptop computers to exhibit the cooling practicality and ideal areas of fans. Tolga Acikalin et.al, (2006)[4] used Design of Experimental analysis (DOE)and came to know that frequency offset, fan amplitude, the separation between the fans and heat source have most prominent effect on cooling ability of fans. Among these, frequency offset plays a major role. Ihara and WatanBe[5]conducted flow visualization experiments and numerical analysis to know the quasi two-dimensional flows around the free finishes of adaptable single plate, two plates both swaying with vast abundancy. Kim. et al,[6] considered stream field developedby a vibrating cantilever plate utilizing phase settled particle image velocimetry (PIV) estimations just as smoke perception procedure.Kimber et.al,[7] examined local heat transfer coefficients for a solitary piezo electric fan at different vibration amplitudes and gaps. They found 2D forms of local convective heat transfer coefficient field changing from lobed shape at little gaps to practically round shapes at medium gaps, and circular shapes at bigger gaps.Linderman et.al,[8] performed experiments with parametric resonant fan clusters so as to watch the reliance of volumetric stream rate in a test channel concerning changes in length, resonant frequency and deflection and they found that stream rates of around 10µl/min are delivered by single fans, while an enhancement in stream rates by 270% where three fans are utilizedin a similar size of flow channel.

Kimber et.al,[9] conducted experiments with arrays of piezo electric fans for determining local heat transfer coefficients and compared the results with single piezo electric fan. The result recommended that a pitch P/A = 1.5 yields the biggest increment in all the zonesof enhanced heat transfer (roughly 15%) where, compared with heat of a single fan. As the pitch ends up becoming smaller or bigger, the relative improvement is found to bediminishing. J.C.

Shyu, et.al,[10] utilizes finger like piezo electric fan with just a single actuator to actuate four adoptable rectangular with sharp edges to improve the cooling of plate balance clusters. They found that, when piezo fan positioned at X/L = 0.5and H= 5mm, the heat transferimprovement scope ofboth of fan array orientations, (horizontal or vertical) are similar (i.e.,1.5-3.3) [10].

In the present work, performance of piezo electric fan is analyzed by using the FLOTHERM 12.1. Piezo electric fan is inserted in electronic circuits besides aheat sink to analyze temperature distribution. piezo fan is kept at different positions with respect to heat sink of pin fin typeand compared the temperature distribution over heat sink for three different spacings between the fins.

FLOTHERM is an advanced CFD tool, used in the present study. FLOTHERM solves airflow and heat transfer problems in electronic components by solving conservation equations within each small grid cell of fluid and solid spaces. FLOTHERM modelling involves numerical solution for Navier-Stokes equations and uses Finite Volume approach. Solution is obtainediteratively, since the equations are non-linear and coupled. It helps to create virtual models of electronic equipment and perform thermal analysis. Its wide applications arein aerospace, defense, automotive and electronics.

FLOTHERM uses multi grid solver optionto solveof linearized governing equations in temperature by using multi grid acceleration, and it also improves convergence and reduces the computational time for the problems of heat transfer compared with other general-purposesolvers. For solving the pressure and temperature linear equations, it uses the segregated conjugate residual iterative model.

Problem Statement

Rate of heat transfer enhancement increases with fins. In present work, piezo fan is using to increase the rate of heat transfer and inserted using FLOTHERM to analyze the best optimal space between the heat sink of the size (20mm*20mm*1mm) with fin size of (1mm*1mm*2mm). Heat sink inserted in between the acrylic block of size (150mm*150mm). 3D and 2D pictures of heat sink with pin fin model and fin arrangements are shown in Fig.1 & Fig.2

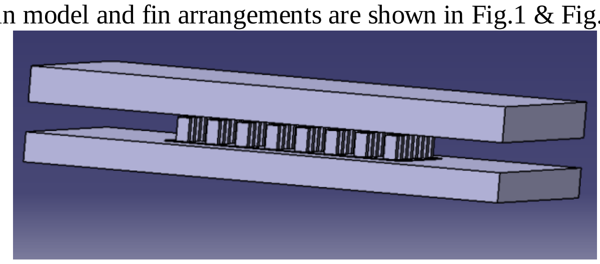


Fig. 1 3D pin fin model

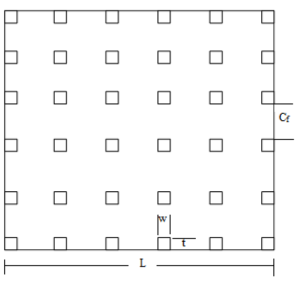


Fig. 2 Heat sink with fins 2D model

II. MATHEMATICAL MODELING

Physical model

Uniform heat was given to the base of the heat sink.The heater set of 150mm*150mm was set into a horizontal acrylic block. With an offset of 5mm, an acrylic block was set above, with the blower having laminar fixed velocity at the central axis of the heat sink, with the flowrate of 750mL/min ($1.25*10^{-5} m^3/s$) and 20^0C inlet air temperature to make allowances for system pressure loss. 1W heat source is applied to heater uniformly[12].

FLOTHERM 12.1used for simulations solves the generic governing equationsby using finite volume approach.

$$\frac{\partial}{\partial t}(\rho\phi) + div(\rho \bar{\nabla} \varphi - \Gamma_{\phi} grad\phi) = S_{\phi}$$

Transient + convection - diffusion = Source

FLOTHERM uses detailed design data which can be automatically imported from mechanical and electrical automation tools thus reducing cycle time to days or overnight against multiple weeks. To know reliable thermal management inPCB(Printed-Circuit-Board)s and IC(Integrated Chips)s, thermal analysis must be done at multiple levels. CFD software connects these levels.

Geometry and thermophysical data

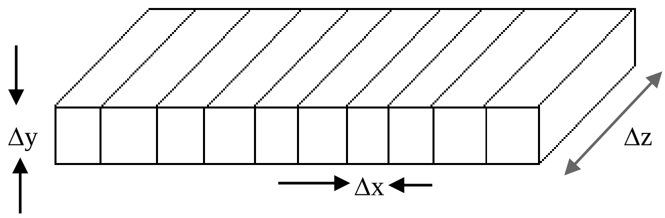
Heat sink – 20mm*20mm*1mm
Fin – 1mm*1mm*2mm
Piezo fan – 0.8mm*0.8mm
Flow rate – $1.25*10^{-5}$ m^3/s(750mL/min)
Inlet air temperature – 20^0C
Heat source – 1W

Solution Methodology

The governing differential equations i.e., mass and momentum conservative equations are discretized and subsequently solved by applying proper boundary conditions. In FLOTHERM there is a clear differentiation between the solid and fluid zones and the appropriate thermo physical properties are assigned to the respective materials and fluid.

Finite Volume Equations

Consider a 1D uniform, transient grid in X-direction



The details of the governing equations and the grid cell nomenclature adopted by FLOTHERM is given as under.

Grid cell nomenclature:

Finite volume equations are derived by volume integration over each grid cell.

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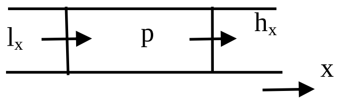


cell volume: $V_p = \delta x \delta y \delta z$
X- direction face area: A_x=δyδz

a) Continuity equation:

$$\iiint_{zyx} \frac{\partial \rho}{\partial t} \mathrm{d}x\, \mathrm{d}y\, \mathrm{d}z = \left(\frac{\rho_P - \rho_t}{\partial t}\right) \delta x \delta y \delta z = \left(\frac{\rho_P - \rho_t}{\delta t}\right) V_p$$

$$\iiint_{zyx} \frac{\partial(\rho u)}{\partial x} dxdydz = \iint_{zy} [(\rho u)_{hxf} - (\rho u)_{lxf}] dydz$$

$$= (\rho u)_{hxf} \delta y \delta z - (\rho u)_{lxf} \delta y \delta z$$

$$= (\rho u)_{hxf} A_x - (\rho u)_{lxf} A_x$$

Complete equation is

$$\left[\frac{\rho_P - \rho_t}{\delta t}\right] V_P + (\rho u)_{hxf} A_x - (\rho u)_{lxf} A_x = 0$$

Rate of increase of mass in cell + Difference between outflow and inflow = 0
for steady state or constant density:
Mass outflow - Mass inflow=0

b) Temperature Equation:

1) Transientterm $\iiint_{\mathbf{zyx}} \frac{\partial(\rho C_p T)}{\partial t} \mathrm{dx\ dy\ dz} =$ $\rho C p T p{-}\rho C p T t \partial t V p$

2) *Convection term*

$$\frac{\partial(\rho u C_p T)}{\partial x} dx\, dy\, dz$$

$$= ii \left[(\rho u C_p T)_{hcf} - (\rho u C_p T)_{lxf}\right] dydz$$

$$= \left[(\rho u C_p T)_{hxf} A_x - (\rho u C_p T)_{lxf} A_x\right]$$

For u is positive

$$T_{hxf} = T_p \rho_{hxf} = \rho_p$$

$T_{lxf} = T_{lx} \rho_{lxf} = \rho_{lx}$(and vice versa)
convection term = $\rho_p C_p u_{hxf} T_p A_x - \rho_{lx} C_p u_{lxf} T_{lx} A_x$

3) *Conduction term*

$$\iiint_{zyx} -\frac{\partial}{\partial x} \lambda \left(\frac{\partial T}{\partial x}\right) dx\, dy\, dz$$

$$= -\iint_{zy} \left[\lambda \left(\frac{\partial T}{\partial T}\right)_{hxf} - \lambda \left(\frac{\partial T}{\partial y}\right)_{lxf}\right] dy\, dz$$

$$= \left[\lambda \left[\frac{T_{hx} - T_p}{\delta x}\right] - \lambda \left(\frac{T_p - T_{lx}}{\delta x}\right)\right] A_x$$

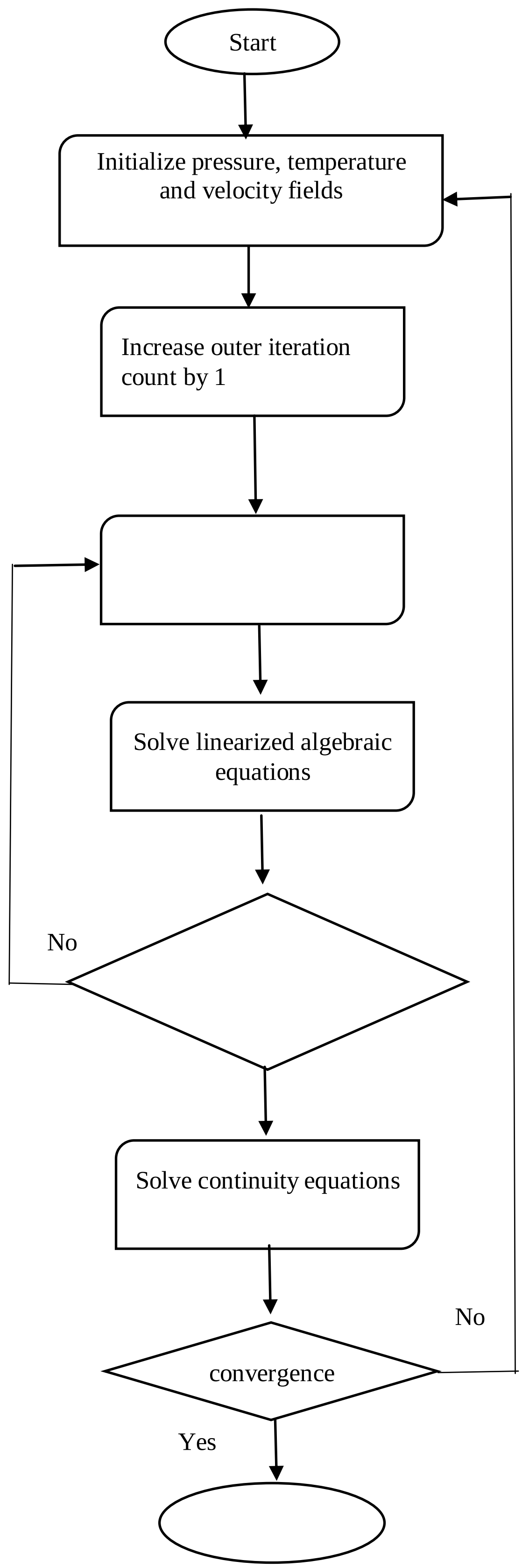
4) *Source term*

$$\iiint_{zyx} S\, dx\, dy\, dz = S_p V_p$$

complete equation is

$$T_p \left[\frac{\rho_p C_p V_p}{\delta t} + \rho_p C_p u_{lxf} A_x + \frac{\lambda A_x}{\delta x} + \frac{\lambda A_x}{\delta x}\right] - \left[\frac{\rho_p C_p V_p T_t}{\delta t} + \left[\rho_p C_p u_{lxf} A_x + \frac{\lambda A_x}{\delta x}\right] T_{lx} + \frac{\lambda A_x}{\delta x} T_{hx}\right] = s$$

Finite Volume Algorithm flowchart



III. RESULTS AND DISCUSSION

2D models of different types of heat sinks

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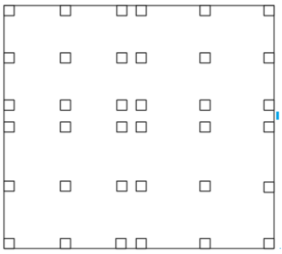
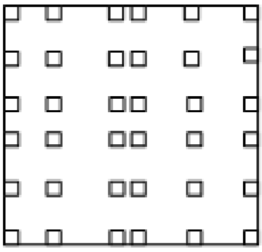


Fig. 3 n_f = 6, C_f = 0.5mm



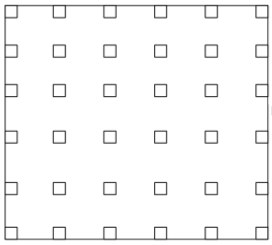


Fig. 4n_f = 6, C_f = 1.5mm Fig. 5 n_f = 6. C_f = 2.4mm

Three different pin fin configurations considered in the present study are shown in Fig. 3, 4&5. An array of 6*6 fins is considered in the study. Three different gaps between the fins i.e., C_f = 0.5, 1.5 and 2.4mm are analyzed. In all three cases, the location of the piezo electric fan is considered to be besides the heat sink but located on the central axis. The velocity and temperature profiles for these three configurations after computational analysis using FLOTHERM are shown in figures 6 & 7 respectively.

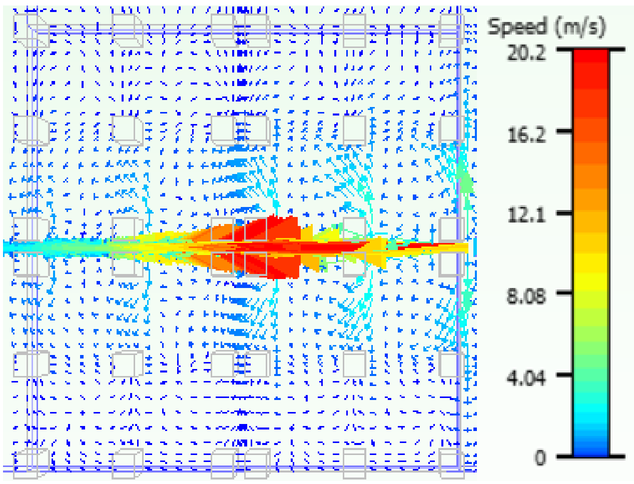


Fig. 6a Velocity contour of n_f = 6, C_f = 0.5mm

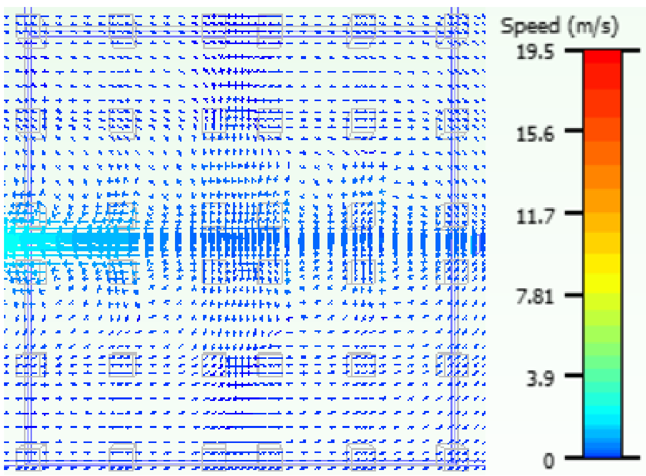


Fig. 6b Velocity contour of n_f = 6, C_f = 1.5mm

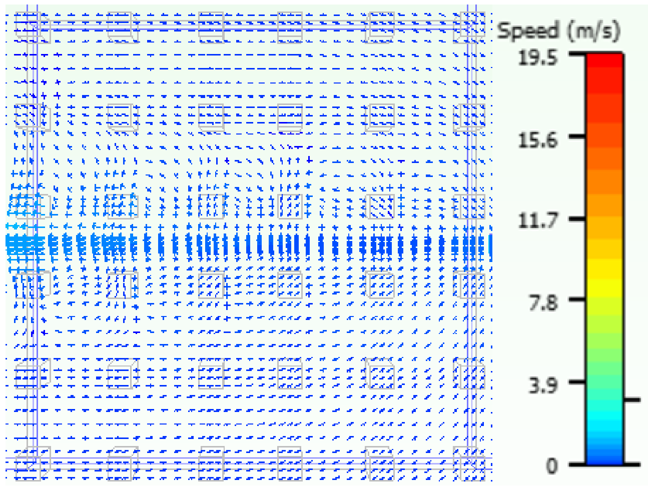


Fig. 6c Velocity contour of n_f = 6, C_f = 2.4mm

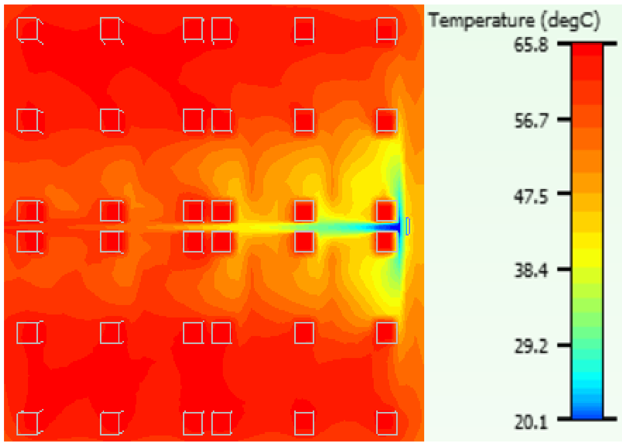


Fig. 7a Temperature contour of n_f = 6, C_f = 0.5mm

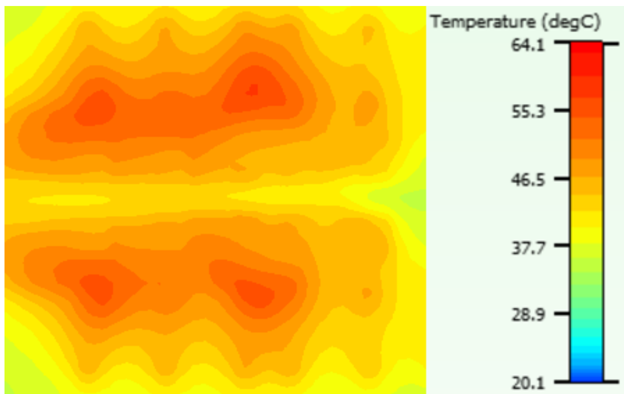


Fig. 7b Temperature contour of n_f = 6, C_f = 1.5mm

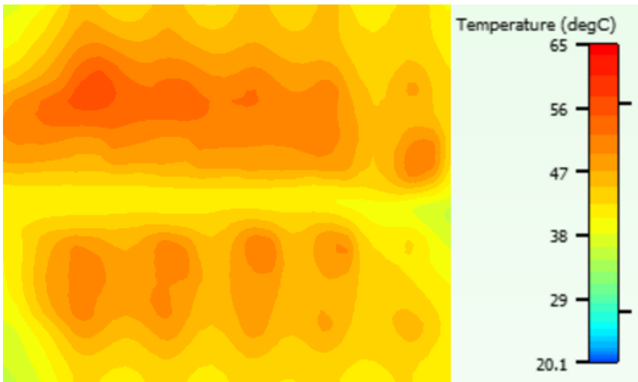


Fig. 7c Temperature contour of n_f = 6, C_f =2.4mm

From Fig.6a, it can be observed that, the maximum velocity is 20.2m/s, while the maximum temperature is 65.8^{0}C.

The second configuration has n_f=6, C_f=1.5mm and upon cooling by alaterally located piezo electric fan and placed on the central axis,realizes a maximum velocity of 19.5m/s and a hotspot temperature of 64.1^{0}C.

Similarly, the maximum velocity and temperature for the third configuration considered in the present study are 19.5m/s and 65^{0}C respectively.

Based on these results, it can be observed that second configuration is giving better results in transporting heat from the heat sink,as it can be seen from the temperature contour plots also. Fig.7b shows temperature distribution also to be consisting more pockets of space occupying less temperature. On the contrary, the first configuration has majority of heat sink area being occupied by highest temperature regions, which is not desirable.

IV. CONCLUSION

Heat sinks with six number of fins are modelled with three different spacings i.e.,0.5,1.5 & 2.4mm with FLOTHERM, an advanced CFD tool. From the results it is concluded that the optimum spacing between fins should be 1.5mm with (6*6)heat sink.It reduces the maximum hotspot temperature from 167^{0}C to 64.1^{0}C with velocity of 19.5m/s.

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