Numerical Analysis on Natural Convection Heat Transfer of a Heat Sink with Cylindrical Pin Fin

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Abstract. As technology advancement progressed in this information age or commonly known as digital age, thermal management has equally improved to keep up with demands from the electronic sector. Hence, heat sink study has become more and more prominent. Natural convection holds advantages since it is maintenance free and has zero power consumption. The purpose of this research is to study the heat transfer performance of heat sink with parametric variations of number and height of pin fin at temperature 308K, 323K, 338K, 353K and 368K. In addition, effect of porosity ranges from 0.524 to 0.960 on thermal resistance was investigated as well. Study found that heat transfer coefficient increases as temperature difference between heat sink and ambient increases. Thermal resistance decreases when porosity increases until it reaches the minimum and subsequently increases. The optimum porosity shown in this study is around 88%.

Introduction

Engineers in information technology (IT) field are devoted to minimize size of devices hence the increase of power consumption and heat generation with the same surface area. The growth of power density proved to be a challenge for thermal management. Therefore, it compelled research and development in order to cope.

One of the most common types of cooling methods practiced in the industry is the used of heat sink with natural convection. Natural convective heat transfer is heat transfer between a surface and a fluid moving over it with the fluid motion caused entirely by the buoyancy forces that arise due to the density changes that result from the temperature variations in the flow [11].

Heat sink is defined as a passive component that cools a device by dissipating heat into the surrounding air [1]. Heat sink is favored in the industry since it can be easily obtained in various size and shape as desired. While free convection is preferred due to its zero power consumption and hence, maintenance free and has high system reliability. A cooling system is designed to enhance the performance of device and not otherwise. Thus, free convection is selected in cooling devices.

Methodology

Assumption

Assumptions are made to simplify calculation and obtain results sooner with the condition that assumptions made would not compromise the quality of end result. In present study, several assumptions listed as below are applied:

i. Flow is laminar and three – dimensional
ii. Steady state heat transfer

iii. Air is treated as ideal gas with constant properties (thermal and fluid) except for density difference with respect to the direction of gravity (Boussinesq approximation)
iv. Temperature across pin fins is constant

Governing Equation

Given that air is the working fluid in this study, there are three law of conservation must be obeyed, namely conservation of mass, conservation of momentum and conservation of energy. These laws are also the governing equations used to solve for natural convection problem.

In present study, Boussinesq approximation is applied to reduce the overall complexity and simplify analysis. Boussinesq suggested in 1903 that all properties of fluid are treated as constant except for density changes in the gravity term of body force in momentum equation [10]. Since the only body force present in natural convection is gravitational force, body force term in momentum equation will only exist in the direction of gravity, which is in y direction in present study.

Continuity Equation:

$$\frac{\partial(u)}{\partial x}+\frac{\partial(v)}{\partial y}+\frac{\partial(w)}{\partial z}=0 \tag{1}$$

Momentum Equation in x, y and z directions:

$$u\frac{\partial(u)}{\partial x}+v\frac{\partial(v)}{\partial y}+v\frac{\partial(v)}{\partial z}=-\frac{\partial P}{\rho\partial y}+v\left(u\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}+\frac{\partial^2 v}{\partial z^2}\right)+\beta g\left(T-T_0\right) \tag{2}$$

$$u\frac{\partial(w)}{\partial x}+v\frac{\partial(w)}{\partial y}+w\frac{\partial(w)}{\partial z}=-\frac{\partial P}{\rho\partial z}+v\left(u\frac{\partial^2 w}{\partial x^2}+\frac{\partial^2 w}{\partial y^2}+\frac{\partial^2 w}{\partial z^2}\right) \tag{3}$$

Energy Equation:

$$u\frac{\partial(T)}{\partial x}+v\frac{\partial(T)}{\partial y}+w\frac{\partial(T)}{\partial z}=a\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2}\right) \tag{4}$$

Computational Fluid Dynamics

Computational fluid dynamics (CFD) process consists of three categories, pre-processing, processing (solver) and post- processing. Pre-processing is where geometry is drawn or modeled in the software and said geometry will be meshed. Given that the heat sink is symmetrical, only a quarter of it is considered for computation in order to save computational time. Control volume of current numerical study is 0.15m × 0.15m × 0.35m (W x L x H) as shown in figure below.

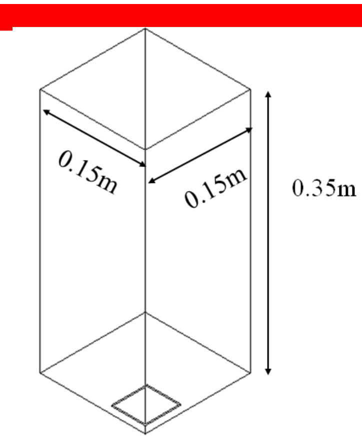

Fig. 2.1 Control Volume

After meshing, geometry is ready for solving process at the processing stage. The purpose of this processing stage is to mirror the real situation as closely as possible. Boundary condition for each surface is assigned as shown in Figure 2.2 and Figure 2.3. Wall boundary condition is treated as a stationary wall that has no slip condition or has zero velocity relative to the boundary while pressure inlet and outlet boundary condition is set at zero gauge pressure and pressure outlet has a backflow temperature of 298K. Convergence criteria for all variables are set at 1 × 10-5 to improve accuracy.

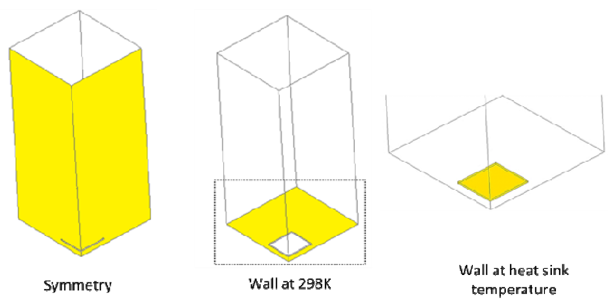

Fig. 2.2 Boundary Conditions

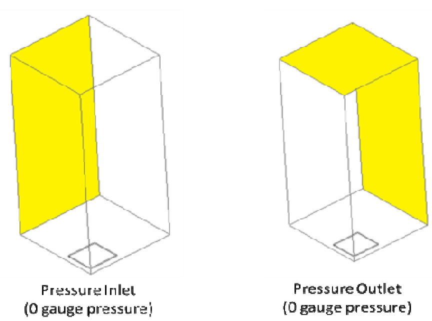

Fig. 2.3 Boundary Conditions

Then, assign solution method and initialize solution before running calculation until it converged. Semi - Implicit Method for Pressure-Linked Equation (SIMPLE) algorithm is used to couple pressure and velocity. Pressure Staggering Option (PRESTO) is chosen for pressure spatial discretization since strong body force present in current study. On the other hand, spatial discretization momentum and energy second order upwind for higher accuracy. The process is repeated for different number of mesh to achieve mesh independence. Process flowchart is illustrated in Figure 2.4.

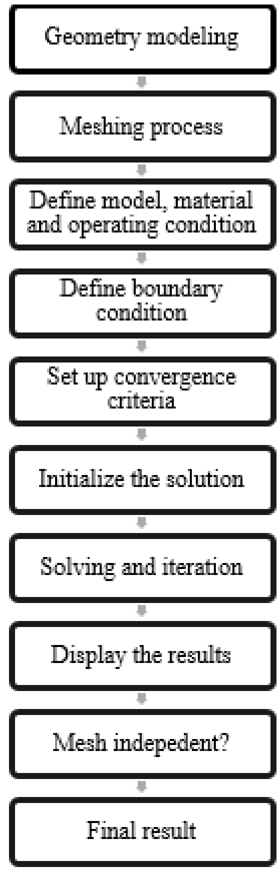

Fig. 2.4 Process Flow Chart

Heat Sink Configuration in This Study

Numerical study was carried out on six pin fin configuration namely Sample 1 to Sample 6 and a flat plate as shown in Figure 2.5. Sample 1 to Sample 4 has the same fin height, H but different fin spacing, S and porosity ϕ. While Sample 5 and Sample 6 have the same fin spacing, and porosity with Sample 2 but different fin height, H. Detail dimension of each Sample is listed in Table 2.1.

	D (mm)	S (mm)	H (mm)	t (mm)	ϕ
Flat Plate	-	-	-	2	-
Sample 1	4	13	10	2	0.960
Sample 2	4	8	10	2	0.881
Sample 3	4	5	10	2	0.759
Sample 4	4	3	10	2	0.524
Sample 5	4	8	15	2	0.881
Sample 6	4	8	20	2	0.881

Table 2.1 Heat Sink Dimension

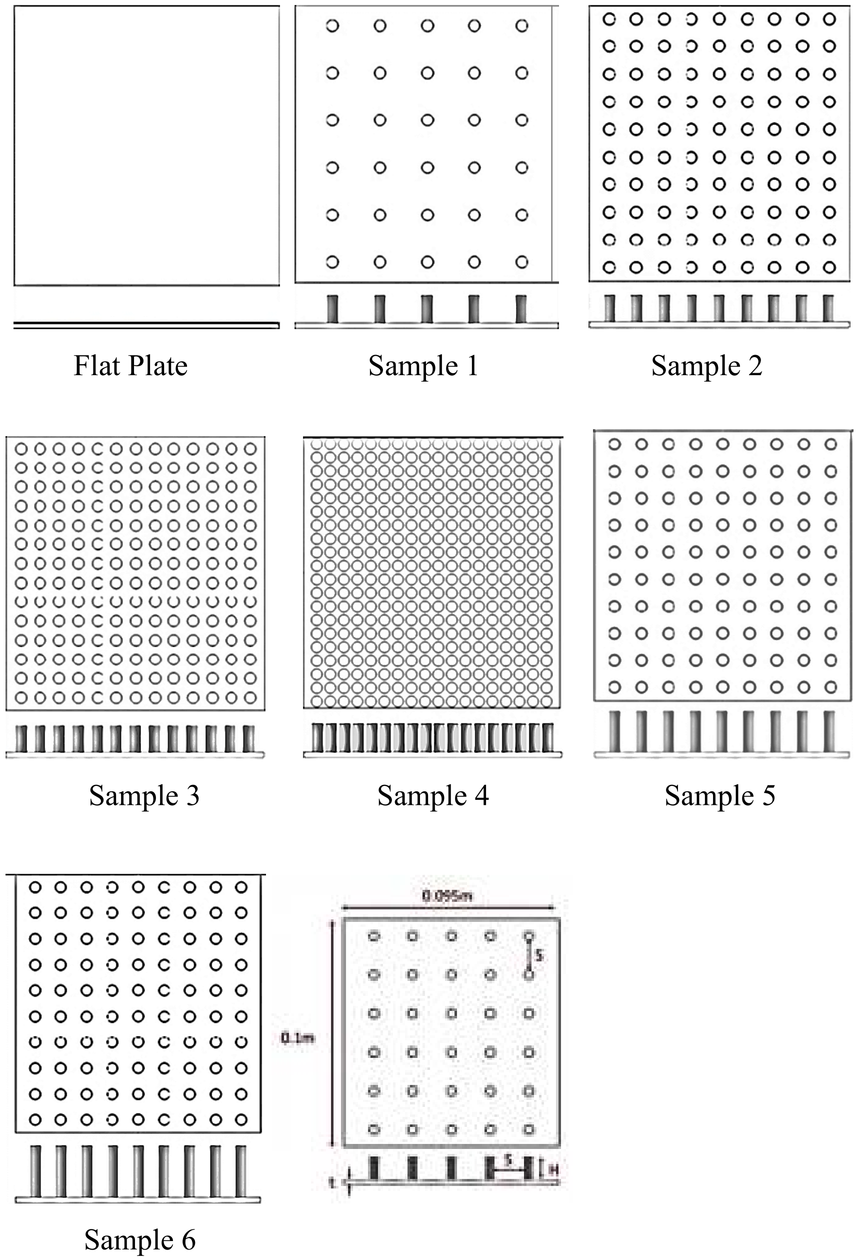

Fig. 2.5 Heat Sink Configuration

Results & Discussion

Validation

Present study was validated against experimental results for natural convection on flat plate by Huang et al. [4]. Validation result is shown in graph of Nusselt number against Rayleigh number in Figure 3.1. Data of experimental study was extracted from graph in journal with the help of software, Enguage.

Rayleigh number was calculated at film temperature, Tf which is the average of flat plate and ambient temperature while Nusselt number was obtained through simulation. Results comparison indicates that simulation was in the correct path given that both studies show the same pattern where Nusselt number increases as Rayleigh number increases.

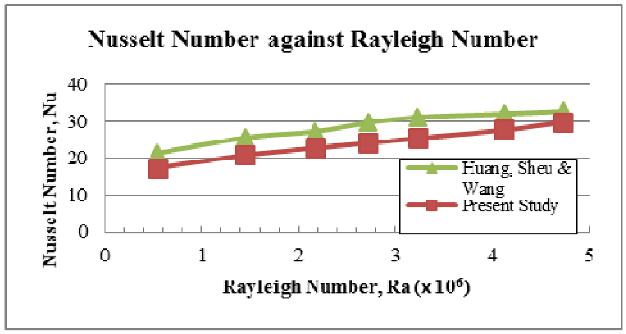

Fig. 3.1 Comparison of present results of flat plate with Huang et al., 2008

Effect of Temperature Difference

Temperature difference plays an important role in heat transfer. Simulation was done on four heat sink configurations with constant height, namely Sample 1 to Sample 4 at five different temperatures which were 308K, 323K, 338K, 353K and 368K. Ambient temperature was fixed at 25°C or 298K. Effect of temperature difference is shown Figure 3.2 Results clearly show that heat transfer coefficient increases with the rise of temperature difference in all heat sink configuration as well as flat plate. This observation coincides with findings of Naidu et al. [8] where they studied the effects of base inclination of fin array on heat transfer.

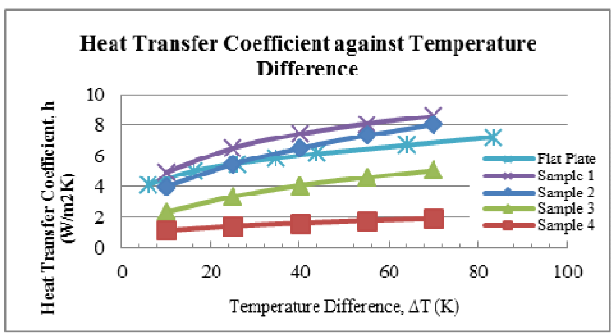

Fig. 3.2 Graph of Heat Transfer Coefficient against Temperature Difference

Density is a function of temperature, therefore higher temperature difference leads to higher density changes. Since buoyance force is directly proportional to density changes, higher temperature difference causes larger buoyance force and enhance heat transfer coefficient.

Figure 3.2 also shows that heat sink does has better heat transfer with the help of extended surface compare to flat plate but it is only up to a certain degree only. Adding more pin fins will only decrease performance of heat transfer since pin fins array will act as blockage of air flow in the process.

Effect of Heat Sink Porosity on Thermal Resistance

Heat sink porosity is the volume fraction of fluid inside heat sink (Vf/Vff) while thermal resistance, R is 1/ (hA). Thermal resistance of Sample 1 to Sample 4 with porosity of 0.960, 0.881, 0.759 and 0.542 respectively was investigated. Figure 4.5 shows that as porosity increases, thermal resistance of heat sink decreases until it reaches minimum and subsequently increases. Current finding has the same trend as experiment results by Huang et al. [4] on square pin fin. The optimum porosity obtained in this study is 0.88 or 88% of air volume in total volume of heat sink.

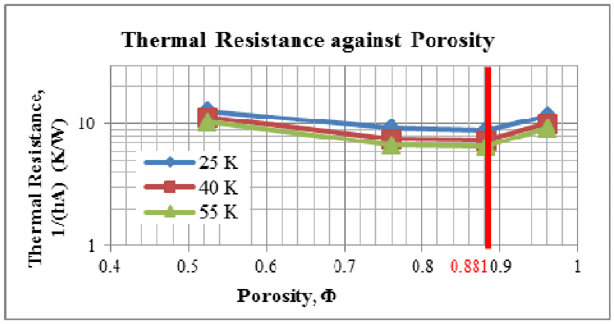

Fig. 3.3 Graph of Thermal Resistance against Porosity

Increase of porosity improves heat transfer performance since it endorses penetration depth of ventilation in heat sink. When air circulation is faster and smoother, hotter air with lower density moves away from the heat sink in higher speed and void is filled with colder air from the surrounding. Temperature difference between heat sink and surrounding is maintained at its peak and thus heat transfer is at its best.

However, when porosity continues to increase, thermal resistance increases after it reaches minimum at 0.881. It should be noted that at porosity equals to 0.960 or also known as Sample 1, heat transfer coefficient is highest compare to the rest of heat sink configuration. Nonetheless, the increase in heat transfer coefficient is insufficient to compensate the reduction of surface area. Hence, thermal resistance at porosity of 0.960 is greater compare to porosity of 0.881.

Effect of Fin Height on Heat Transfer Performance and Thermal Resistance

Heat sink configuration, Sample 2 with the optimal porosity at 0.881 was chosen to carry out study on effects of fin height on heat transfer performance and thermal resistance. Three different height was analyzed which are 10 mm, 15mm and 20mm. Figure 3.4 shows the effect of fin height on heat transfer coefficient at the same fin spacing of 8mm and 5 temperature differences. Figure 3.4 clearly shows that heat transfer coefficient not only increases with temperature difference as established earlier, it also increases with increases of fin height. Higher fin height also means more surface area in contact with surrounding to transfer heat. Hence, heat transfer performance improves with addition of surface area.

Since taller fin has higher heat transfer coefficient and surface area, it is no doubt that taller fin will have lower thermal resistance and it had been proven in Figure 3.5 According to Figure 3.5, as fin height decreases, thermal resistance decreases for all temperature difference.

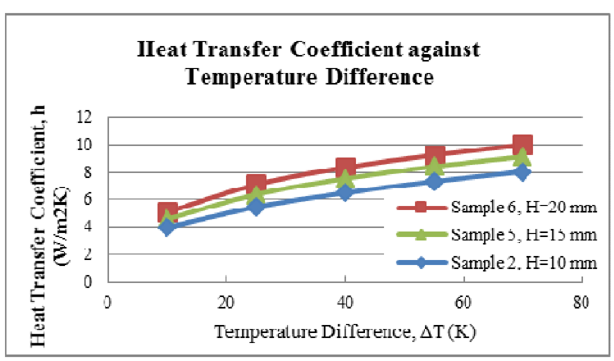

Fig. 3.4 Graph of Heat Transfer Coefficient against Temperature Difference

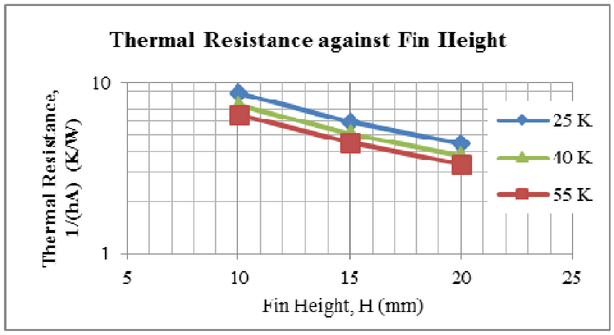

Fig. 3.5 Graph of Thermal Resistance against Fin Height

Conclusion

The objectives of this study has been successfully achieved by using CFD software ANSYS - Fluent. As temperature difference increases, heat transfer coefficient of heat sink increases in all cases, regardless of heat sink configuration. Study found that heat sink could improve heat transfer performance by adding extended surface area in contact with colder air in the surrounding. However, further increment of fin number deteriorates performance.

As heat sink porosity increases, thermal resistance decreases until it reaches minimum and subsequently increase. The lowest thermal resistance indicates optimum heat sink porosity, which is 0.881. In addition to fin spacing, fin height also plays a role in heat transfer performance. While results of this study are by no means definitive, findings shows that the taller the pin fin, the higher the heat transfer coefficient and at the same time, reduces thermal resistance. Hence, encourage the use of taller pin fin.

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