Penerbit Akademia Baru

Journal of Advanced Research in Fluid Mechanics and Thermal Sciences
ISSN (online): 2289-7879 | Vol. 2, No. 1. Pages 13-22, 2014

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

Y. Z. Qin[a], *A. N. Darus*[b] *and N. A. Che Sidik*[*,c]

Faculty of Mechanical Engineering, Universiti Teknologi Malaysia, 81310 Skudai, Johor, Malaysia

[a]ziqin@gmail.com, [b]amer@fkm.utm.my,[,c]azwadi@fkm.utm.my*

Abstract – *As technology advancement progresses 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 the temperatures of 308K, 323K, 338K, 353K and 368K. In addition, the effect of porosity ranging from 0.524 to 0.960 on thermal resistance was investigated as well. The study found that heat transfer coefficient increases as the temperature difference between heat sink and ambient increases. Thermal resistance decreases when the porosity increases until it reaches the minimum and subsequently increases. The optimum porosity shown in this study is around 88%.*

Keywords: Natural Convection, Cylindrical Pin Fin, Heat Sink, Porosity

1.0 INTRODUCTION

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

One of the most common types of cooling methods practiced in the industry is the use of heat sink with natural convection. Natural convective heat transfer is a heat transfer between a surface and a fluid moving over the surface, with the fluid motion is entirely caused by the buoyancy forces that arise due to the density changes as a result of temperature variations in the flow [1].

Heat sink is defined as a passive component that cools a device by dissipating heat into the surrounding air [2]. Heat sink is favoured in the industry since it can be easily obtained in various size and shape as desired. Meanwhile, free convection is preferred due to its zero power consumption and hence, it is 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 [3-5].

2.0 METHODOLOGY

Journal of Advanced Research in Fluid Mechanics and Thermal Sciences
ISSN (online): 2289-7879 | Vol. 2, No. 1. Pages 13-22, 2014

2.1 Assumption

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

i. Flow is laminar and three dimensional

ii. Steady state heat transfer

iii. Air is treated as an ideal gas with constant properties (thermal and fluid), except for different density with respect to the direction of gravity (Boussinesq approximation)

iv. Temperature across pin fins is constant

2.2 Governing equation

Given that air is the working fluid in this study, there are three laws 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 natural convection problem.

In the present study, Boussinesq approximation was 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 [6-8]. Since the only body force present in natural convection is gravitational force, thus the body force term in momentum equation will only exist in the direction of gravity, which is in y direction in the present study [9-13].

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:

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

2.3 Computational fluid dynamics

Journal of Advanced Research in Fluid Mechanics and Thermal Sciences
ISSN (online): 2289-7879 I Vol. 2, No. 1. Pages 13-22, 2014

Computational fluid dynamics (CFD) process [3] consists of three categories; pre-processing, processing (solver) and post-processing. Pre-processing is where the geometry is drawn or modeled in the software and the said geometry will be meshed. Given that the heat sink is symmetrical, only a quarter of it was considered for computation in order to save computational time. The control volume of the current numerical study is 0.15 m × 0.15 m × 0.35 m (W x L x H) as shown in the figure below.

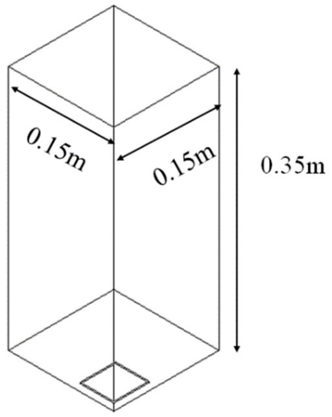


Figure 1: Control volume

After meshing, the 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 was assigned as shown in Figure 2 and Figure 3. Wall boundary condition was 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 were set at zero gauge pressure and pressure outlet had a backflow temperature of 298K. The convergence criteria for all variables were set at 1×10^{-5} to improve accuracy.

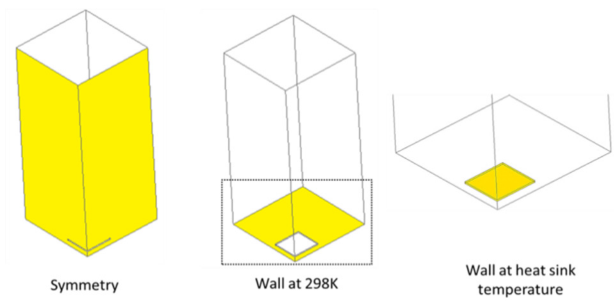
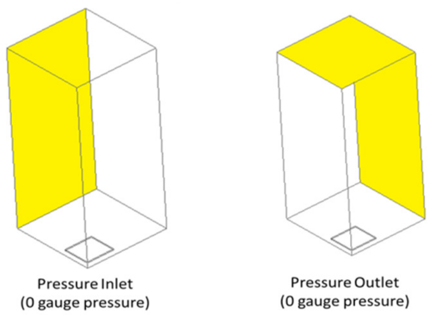


Figure 2: Boundary conditions



Journal of Advanced Research in Fluid Mechanics and Thermal Sciences
ISSN (online): 2289-7879 I Vol. 2, No. 1. Pages 13-22, 2014

Figure 3: Boundary conditions

Then, solution method was assigned and solution was initialized prior to calculation until it converged. Semi-Implicit Method for Pressure-Linked Equation (SIMPLE) algorithm was used to couple pressure and velocity. Pressure Staggering Option (PRESTO) was chosen for pressure spatial discretization due to the presence of strong body force in the current study. On the other hand, spatial discretization momentum and energy second order upwind were used for higher accuracy. The process was repeated for different number of mesh to achieve mesh independence. The process flow chart is illustrated in Figure 4.

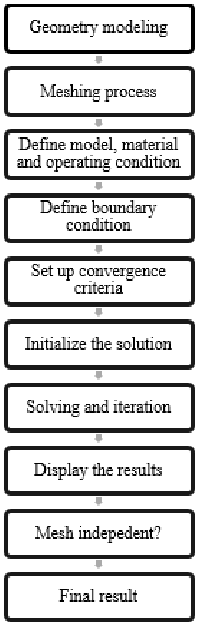
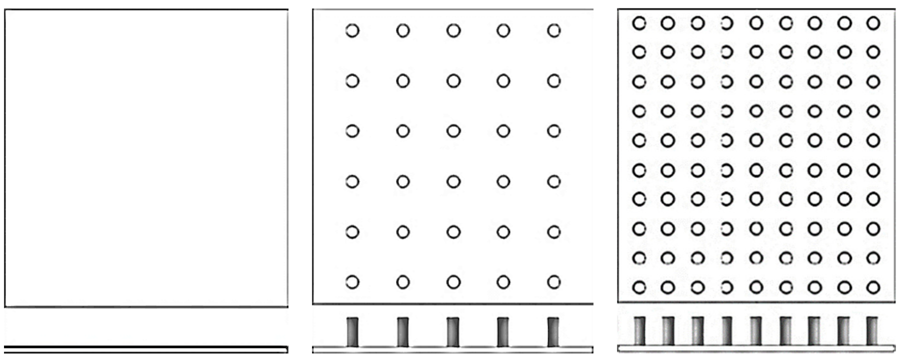


Figure 4: Process flow chart

2.4 Heat sink configuration

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



Journal of Advanced Research in Fluid Mechanics and Thermal Sciences
ISSN (online): 2289-7879 I Vol. 2, No. 1. Pages 13-22, 2014

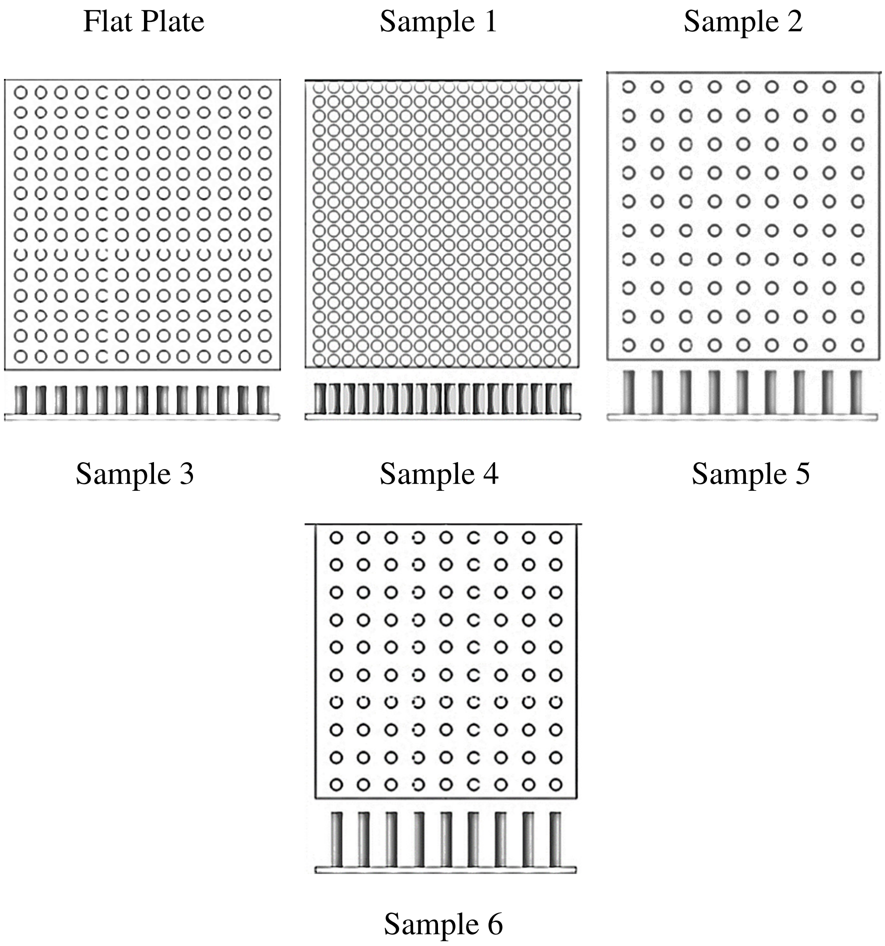


Figure 5: Heat sink configuration

Table 1: Heat sink dimension

	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

3.0 RESULTS AND DISCUSSION

3.1 Validation

The present study was validated against the experimental results for natural convection on flat plate by Huang et al. [4]. The validation result is shown in graph of Nusselt number against Rayleigh number in Figure 6. The data of experimental study was extracted from the graph in the journal with the help of a software, Enguage.

Journal of Advanced Research in Fluid Mechanics and Thermal Sciences
ISSN (online): 2289-7879 | Vol. 2, No. 1. Pages 13-22, 2014

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

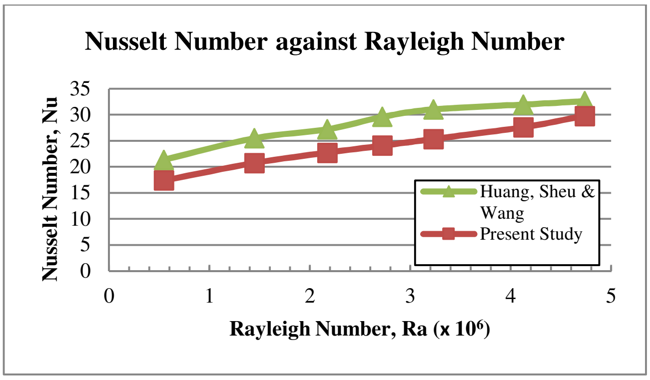


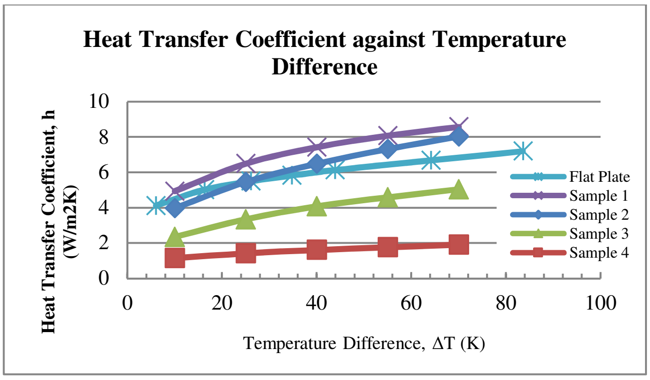
Figure 6: Comparison of results of flat plate from the present study with Huang et al. [4]

3.2 Effect of temperature difference

Temperature difference plays an important role in heat transfer. A 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. The effect of temperature difference is shown Figure 7. The results clearly show that heat transfer coefficient increases with the rise of temperature difference in all heat sink configurations, as well as flat plate. This observation coincides with the findings of Naidu et al. [8], where they studied the effects of base inclination of fin array on heat transfer.

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

Figure 7 also shows that heat sink does has better heat transfer with the help of extended surface compared to flat plate but it is only up to a certain degree only. Adding more pin fins will only decrease the performance of heat transfer since pin fins array will act as blockage of air flow in the process.



Journal of Advanced Research in Fluid Mechanics and Thermal Sciences
ISSN (online): 2289-7879 I Vol. 2, No. 1. Pages 13-22, 2014

Figure 7: Graph of heat transfer coefficient against temperature difference

3.3 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. The current finding has the same trend as the experimental 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 the total volume of heat sink.

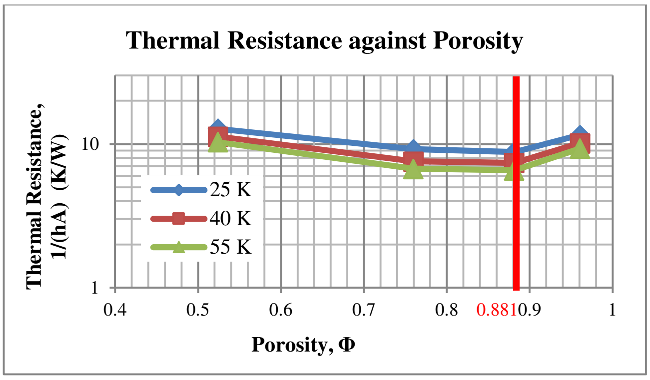


Figure 8: Graph of thermal resistance against porosity

The increase of porosity improves heat transfer performance since it endorses depth penetration 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. The 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 the highest compared to the rest of heat sink configurations. 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 compared to porosity of 0.881.

3.4 Effect of fin height on heat transfer performance and thermal resistance

Heat sink configuration of Sample 2 with the optimal porosity at 0.881 was chosen to carry out study on the effects of fin height on heat transfer performance and thermal resistance. Three different heights were analyzed; 10 mm, 15 mm and 20 mm. Figure 9 shows the effect of fin height on heat transfer coefficient at the same fin spacing of 8 mm and 5 temperature differences. The figure 9 clearly shows that heat transfer coefficient not only increases with temperature difference as established earlier, it also increases with the increase of fin height. Higher fin height also means more surface area in contact with surrounding to transfer heat. Hence, heat transfer performance improves with the addition of surface area.

Penerbit Akademia Baru
Journal of Advanced Research in Fluid Mechanics and Thermal Sciences
ISSN (online): 2289-7879 | Vol. 2, No. 1. Pages 13-22, 2014

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 has been proven in Figure 9. According to Figure 9, as fin height decreases, thermal resistance decreases for all temperature difference.

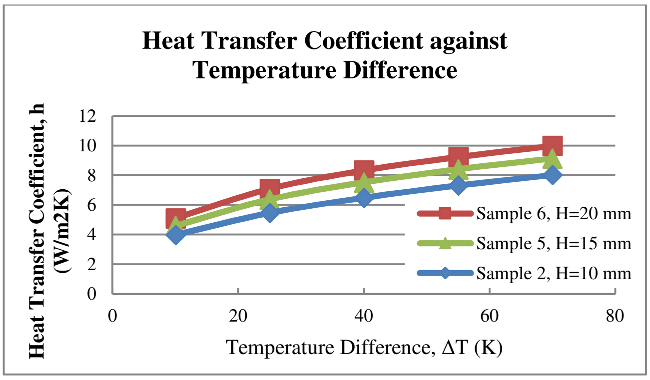


Figure 9: Graph of heat transfer coefficient against temperature difference

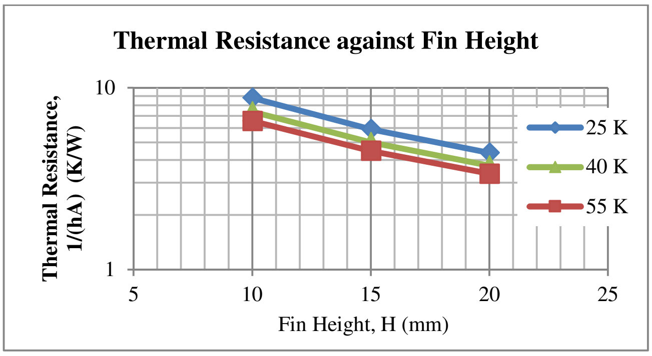


Figure 10: Graph of thermal resistance against fin height

4.0 CONCLUSION

The objectives of this study have 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 configurations. This 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 increases. 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 the results of this study are by no means definitive, the findings show that the taller the pin fin, the higher the heat transfer coefficient and at the same time, reduces thermal resistance. Hence, this encourages the use of taller pin fin.

ACKNOWLEDGEMENT

The authors would like to thank Universiti Teknologi Malaysia and the Ministry of Education of Malaysia for supporting this research activity. This research was financially supported by the Research University Grant 06H23.

REFERENCES

[1] P.H. Oosthuszen, D. Naylor, Introduction to Convective Heat Transfer Analysis, Singapore: McGraw Hill. 1999.

[2] D. Systemes, Convection Heat Coefficient, Retrieved on October http://help.solidworks.com/2012/English/SolidWorks/cworks/Convection_Heat_Coefficient.htm. 2012.

[3] Fluent 6.3 User's Guide. (2006). Fluent 6.3 User's Guide. Retrieved on December from http://cdlab2.fluid.tuwien.ac.at/LEHRE/TURB/Fluent, Inc/fluent6.3.26/help/html/ug/node1013.htm. 2012.

[4] R.T. Huang, W.J. Sheu, C.C. Wang, Orientation effect on natural convective performance of square pin fin heat sinks, International Journal of Heat and Mass Transfer 51 (2008) 2368-2376.

[5] F.P. Incropera, D.P. Dewitt, T.L. Bergman, A.S. Lavine, Foundations of Heat Transfer, (6th ed.). Asia: John Wiley & Sons. 2013.

[6] I. Kamarulzaman, Numerical analysis on thermal performance of staggered pin-fin assembly, Master Thesis, Universiti Teknologi Malaysia, Skudai. 2007.

[7] H.S. Kou, J.J. Lee, C.Y. Lai, Thermal analysis and optimum fin length of a heat sink, Heat Transfer Engineering 24 (2003) 18-29.

[8] S.V. Naidu, V.R. Dharma, B. GovindaRao, A. Sombabu, B. Sreenivasulu, Natural convection heat transfer from fin arrays-experimental and theoretical study on effect of inclination of base on heat transfer, ARPN Journal of Engineering and Applied Sciences 5 (2010) 7-15.

[9] D.A. Nield, A. Bejan, Convection in Porous Media, New York: Springer. 2006.

[10] D.Y. Shang, Free Convection Film Flows and Heat Transfer, New York: Springer Verlag. 2006.

[11] E.A.M. Elshafei, Natural convection heat transfer from a heat sink with hollow/perforated circular pin fins, Energy 35 (2010) 2870-2877.

[12] O.N. Sara, Performance analysis of rectangular ducts with staggered square pin fins, Energy Conversion and Management 44 (2003) 1787–1803.

Journal of Advanced Research in Fluid Mechanics and Thermal Sciences
ISSN (online): 2289-7879 | Vol. 2, No. 1. Pages 13-22, 2014

[13] E.M. Sparrow, S.B. Vemuri, Orientation effects on natural convection/radiation heat transfer from pin-fin arrays, International Journal of Heat and Mass Transfer 29 (1986) 359-368