Comparative Study of Performance of Pin Fin under Forced Convection Heat Transfer

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ABSTRACT

In this study experimental based model for pin fin heat sinks in forced convection has been developed. Key empirical data and correlations for thermal resistance and heat transfer in in-line arrays of circular pin fin arrays were validated through CFD simulations. With same design parameters, such as height diameter and spacing of pin-fin are validate by CFD analysis. The thermal resistance and heat transfer are considered as the multiple thermal performancecharacteristics. Pin fin heat sink having 6×8 array of circular pin. Total number of pin fin is restricted to be 48 and velocity of air flow is set to 2.5 m/s.The heat sinks were constructed of aluminum (thermal conductivity of 209 W/mK) and consisted of an array of staggered pin fins The base of Pin fin heat sink is subjected to a heat load of 40 W. Heat sinks had constant fin height of 60mm,pin diameter of 6 mm and the longitudinal pitch was 10 mm, varied transverse pitch was 8,12 mm.Heatfluxwas applied to the wall using heaters.

Keywords: forced convection, Extended surfaces, pin fin, CFD

INTRODUCTION

Air cooled heat sinks are the workhorse device for all sorts of components in the electronics industry. The widespread availability of user-friendlyComputational fluid dynamics (CFD) software now enablesHeat sink designers and to obtain a reasonably accurate prediction of Heat sink performance. In recent years, the heat sink has been extensively used to provide cooling function for electronics components because the circuit density and power dissipation of integrated circuit chips are rapidlyincreasingin order to increase theheat flux levels within these chips. The transferof large amount of heat flux can create considerable quantities of heat generatedin chips, substrate, and its package. Therefore, it is necessary for employing effective heat sink module to maintain the operating temperature of electronic components at a satisfactory level. If there is appropriate and effective heat sink design, it will critically affect the reliability and life span of chip function. There have been many investigations of the optimum design parameters and selection of heat sink with a high-performance heat removal characteristic(2–

8).**Ellison**[2]and**KrausandBar Cohen**[3] have presented the fundamentals of heat transfer and hydrodynamics characteristics of heat sinks including the fin efficiency, forced convective correlations, applications in heat sinks,etc.**Iyengarand Bar Cohen**[4]determined the least-energy optimization of plate fin heat sinks in the status of forced convection. **Park et al.**[5,6]performed an investigation of numerical shape optimization for high performance of a heat sink with pin-fins. **Park and Moon**[7]proposed the progressive quadratic responsesurface model to obtain the optimal values of design variables for a plate-fin type heat sink.**Sahin et al**. [8] investigated the effect of design parameters on the heat transfer and pressure dropcharacteristics of a heatexchanger using the Taguchi experimental design method. From the above descriptive analysis, the optimal design and selection of effective heat sink module is becoming one of the primary challenges of the computer science and technology industry. In this study, the optimal values of designing parameters of a pin-fin type heat sink (PFHS) are numerically acquired using the quadratic model of response surface methodology (RSM), associated with a sequential approximation optimization (SAO) method to reach the high thermal performance (or cooling efficiency). The RSM relates to the regression analysis and the statistical design of experiments for constructing the global optimization [9] and is one of the most widely used methods to solve the optimization problem in the manufacturing environments [10–13]. To achieve the high thermal performance (or cooling efficiency) under the given design constrain, the predictive model for thermal performance characteristics will be created using the RSM.

[14]**SukhvinderKang**,**MauriceHolahan**presents a physics based analytical model to predict the thermal behavior of pin fin heat sinks in transverse forced flow. Thekey feature of the model is the recognition that unlike plate fins, streamwise conduction does not occur in pin fin heat sinks. Thus, the heat transfer from each fin depends on its local air temperature or adiabatic temperature and the local adiabatic heat transfer coefficient[15]**KoTachiangandFu Ping Chang** has developed an effective procedure of response surface methodology (RSM) forfinding optimal values of designing parameters ofa pin fin type heat sink (PFHS) under constrains of mass and space limitations to achieve the high thermal performance (or cooling efficiency).various design parameters such as height and diameter of pin fin and width of pitch between fins are explored by experiment. The Thermal resistance and pressure drop are considered as the multiple thermal performancecharacteristics.

METHODOLOGY

In this study, the thermal performances of Pin-fins has been investigated by the aid of an academic CFD program, FLUENT 14&the results are compared with those obtained experimentally & numerically for pin fins on same base. For analyzing the above mentioned matter, 3-dimentional flow is considered with physical properties, walltemperature more than the entrance forced air flow temperature. Air flows with velocity of 2.5m/s.Heat transfer rate changes and temperature of fin changes due to air velocity. The equation of conservation of mass, momentum and the energy for stable and flow is described follow:

Continuity equation:

Equation 1 $$\frac{\partial(\rho u_i)}{\partial x_i} = 0.$$

Momentum equation:

Equation 2

$$\frac{\partial}{\partial x_i}(\rho u_i u_j) = \frac{\partial}{\partial x_i}\left(\mu \frac{\partial u_j}{\partial x_i}\right) - \frac{\partial p}{\partial x_j}.$$

Energy equation:

Equation 2 $$\frac{\partial}{\partial x_i}(\rho u_i T) = \frac{\partial}{\partial x_i}\left(\frac{k}{C_p}\frac{\partial u_j}{\partial x_i}\right).$$

Momentum equation (2) and energy equation (3) should be solved simultaneously because velocity and temperature are both unknown, since temperature depends on the velocity and vice versa. Governing equations are solved using a finite volume approach. The convectiveterms are discretized using the power-law scheme, whereas for diffusive terms the central difference is employed. Coupling between the velocity and pressure is made with SIMPLE algorithm. The resultant system of discretized linear algebraic equations is solved with an alternating direction implicit scheme.The procedure for solving the problem is:

i) Create the geometry.

ii) Mesh the domain.

iii) Set the material properties and boundary conditions.

iv) Obtaining the solution

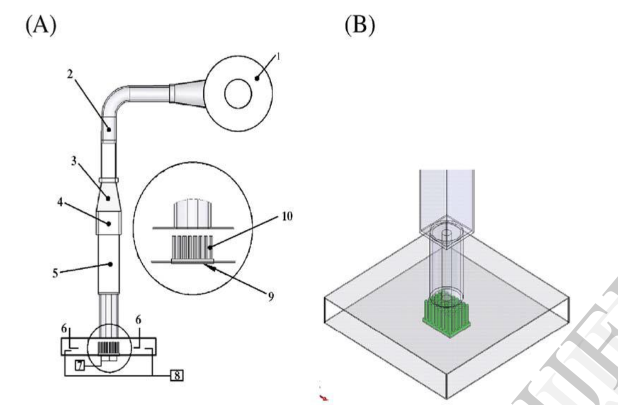
Modeling software UNI-Graphics NX-7.5 creates the geometry and the geometry is imported to the Ansys workbench 14.0 where meshing is done, and exports the mesh to FLUENT. The boundary conditions, material properties, and surrounding properties are set through parameterizedcase files. FLUENT solves the problem until either the convergence limit is met, or the number of iterations specified by the user is achieved.

2. Experimental detail

Experimental apparatus:

The experimental setup under considerationfor optimization is a fan-drive heat sink with pin fins consisting of airBlower, pre-heater, adjustable contraction zone, honeycomb, air flow channel, test section, and measurementfacilities and is illustrated schematically inFig.Air is supplied by a centrifugal blow with the variable-speed drive and then passed through an insulated chamber. The air-flow channel is designed to simulate the electric micro-fan (an external wingdiameterof 65 mm and a motor diameter of 25 mm) installed over the heat sink. The static pressure in air flow channel is measured using a static-pressure tapping located within the middle of this channel. The test section in Fig. 1(B) is constructed by a hollow rectangular block (720×720×150 mm) made up of upper and bottom plate[K.-T. Chiang, F.-P. Chang / International Communications in Heat and Mass Transfer

33 (2006) 836–845]. Air can exhaust through the horizontalaround of test section. The outlet temperature of the air stream located in the horizontal around of test section is measured with thermocouple and data acquisition system. The static pressure tapping measures the static pressure of air-stream exhausting from the horizontal around of test section. The heating unit located in the middle of bottom plate



1. Blower	6. Pitot tube
2. Preheat	7. Hot-wire sensor
3. Chamber	8. Thermocouple sensor
4. Adjustable contraction zone	9. Heating unit (DC power supply)
5. Honeycomb	10. Heat sink

Fig.1 Schematic display of the experimental setup.

The schematic of the commercial extruded PFHS having a 6×8 array of circular pin isstudied in this experiment, and is shown in Fig. 2. Its material is the aluminium alloy 6063-T5 (thermal conductivity 209 W/m K). The PFHS has80×60×5-mm base size which is suitably mounted on the CPU board. The tested heat sink is mounted on the heatingunit with two small tapped holes in the base of the heat sink. The base of PFHS is subjected to a heat load of 40W. The consists of theelectric heater, voltagetransformer, a firebrick of 25mmthickness and the thermal insulator. The electric heater and voltage transformer are used to control the heat flux along the bottom of base plate. The heat generated by heating unit is conductedthrough the heat sink at first and then it is diffused to the environment by means of forced convection.

steady state temperature of PPF heat sink base is measured by using the 25-gauge (0.12 mm diameter wire) copper–constantan thermocouple installed on the base plate. The date of temperature are scanned instantaneously by using adata acquisition and calibrated to within ±0.1 °C.

Fig. 1.A schematic display of the experimental setup.

Fig. 2.

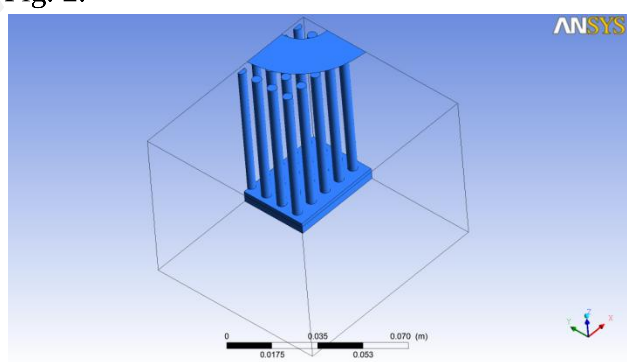


Fig. 2.Designed geometry and dimensions of pin-fin heat sink (PFHS).

Measuring calculation and performance characteristic the thermal performance of heat sink is determined by the heat transfer rate and the capacity of a fan. For a givenoperating condition of a fan, increasing the heat transfer rate is a principal cause of high thermal performance (orcooling efficiency).

Table 1
Design of experimental matrix and results for the PFHS performance characteristics

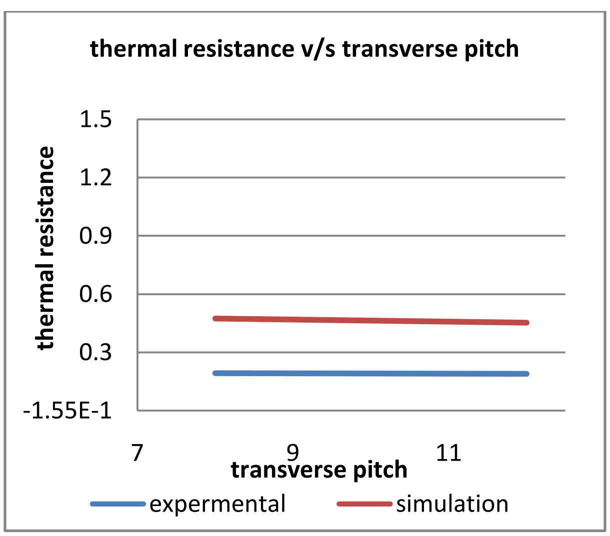
Exp no.	**Design parameters**					**Experimental results**	**CFD Results**
	Fin height A	**Pin diameter B**	**Longitudinal pitch C**	**Transverse pitch D**	**Heat Load (W)**	**Thermal resistance Rt h (°C/W)**	**Thermal resistance Rt h (°C/W)**
1.	**60**	**6**	**10**	**8**	**40**	**.193**	0.475052
2.	**60**	**6**	**10**	**12**	**40**	**.189**	0.452928

4. **Results and discussion:**

We have studied the experimental results and the work is simulated for validation by CFD. The cases are discussed earlier.

<u>Validation of publish experimentalresults with variable Transverse pitch of pin fin</u>

The thermal resistances of pin fin arrays are plotted as a function of transverse pitch input for fin diameter of 6 mm for fin height of 60 mm and for constant longitudinal pitch of 10 mm. It can be seen from figure that thermal resistance of pin fin increases with transverse pitch of pin fin input & for a given variable pitch input, this thermal resistance is more for pin fins as compared to the experimental thermal resistance.we have taken pin surface smooth and the experimental work is based on the roughness of the pin fin surface.



Comparison of thermal resistance with transverse pitch of pin fin

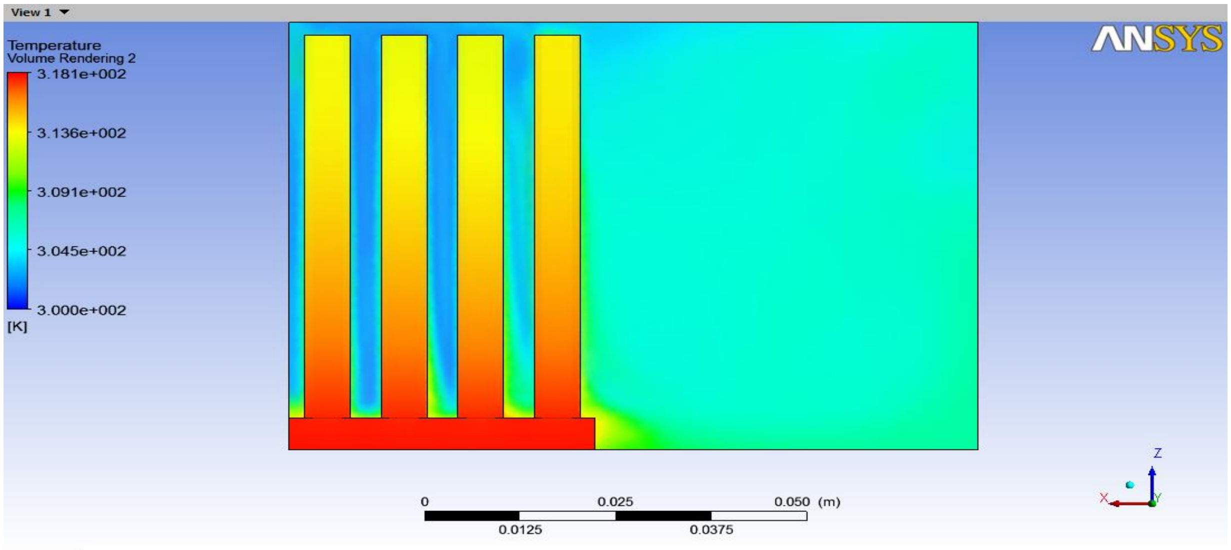


Fig.1 Temperature distribution in Pin fin heat sink with 2.5m/s velocity

This fig. shows temperature distribution in pin fin with constant transverse pitch of 8 mm.as fin base has high temperature of 318.K.Air temperature is surrounding is 300 K.When air flows, due to convection heat transfer rate increases and temperature of air gets increases. In upper portion it is near to 313.6 K.

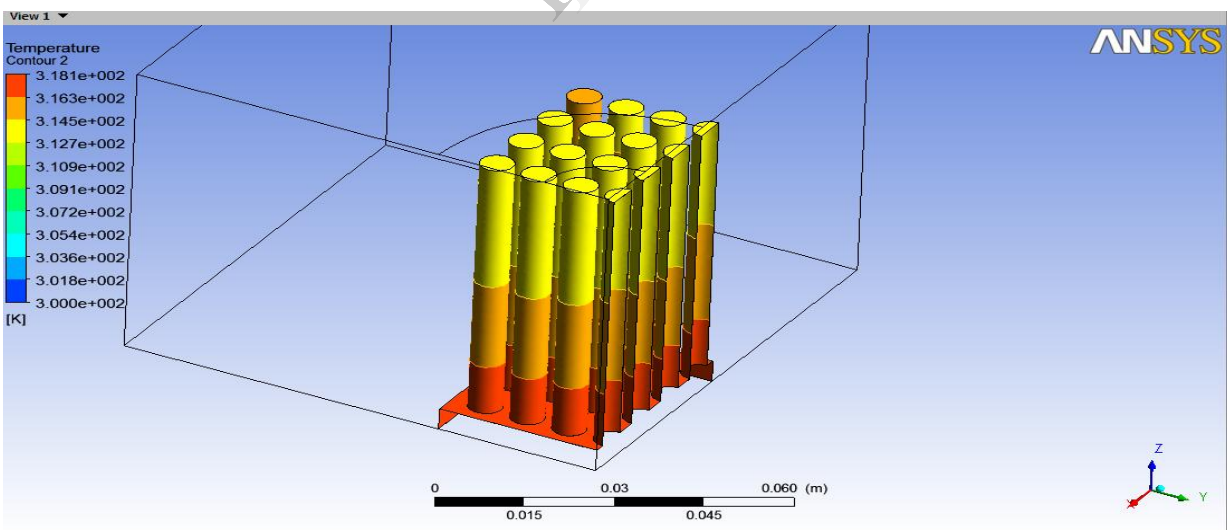


Fig.2 Temperature distribution in Pin fin heat sink with 2.5 m/s velocity

This fig. shows temperature distribution in pin fin with constant transverse pitch of 8 mm. as fin base has high temperature of 318.1K.Air temperature is 300 K.When air flows, due to convection *heat transfer rate increases and temperature of pin fin get decreases. Because of fin spacing some inner fins have higher temperature up to 316.3 K.In upper portion it is near to 314.5 K.*

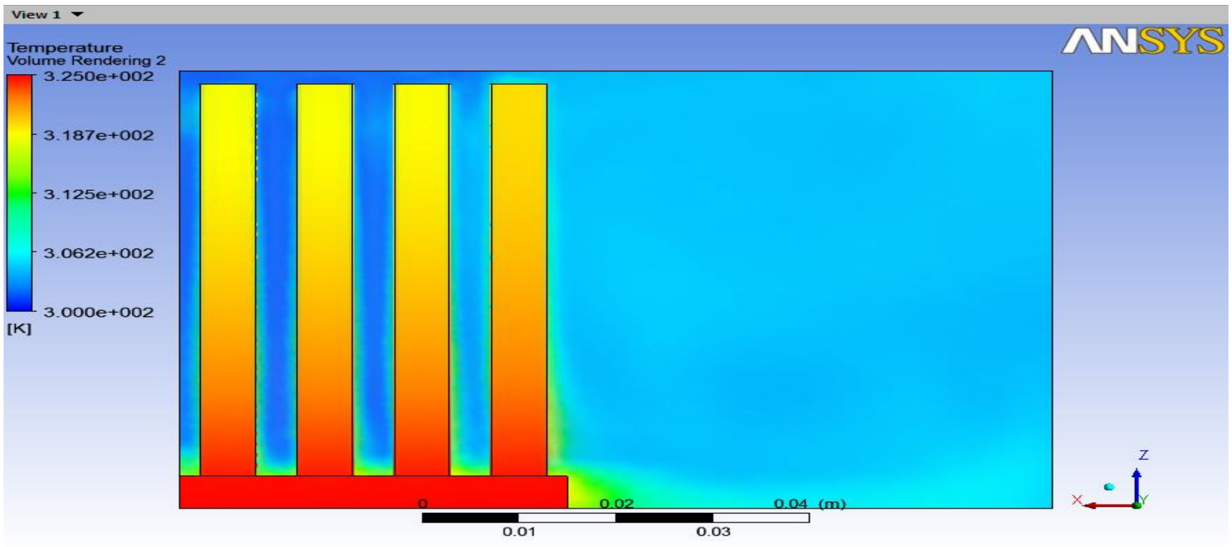


Fig.3 Temperature distribution in Pin fin heat sink with 2.5 m/s velocity

T*his fig. shows temperature distribution of air in pin fin. with constant transverse pitch of 12 mm. as fin base has high temperature of 325.K.Air temperature is 300K in surrounding. When air flows, due to convection heat transfer rate increases and temperature of pin fin get decreases. In upper portion it is near to 318.7 K.*

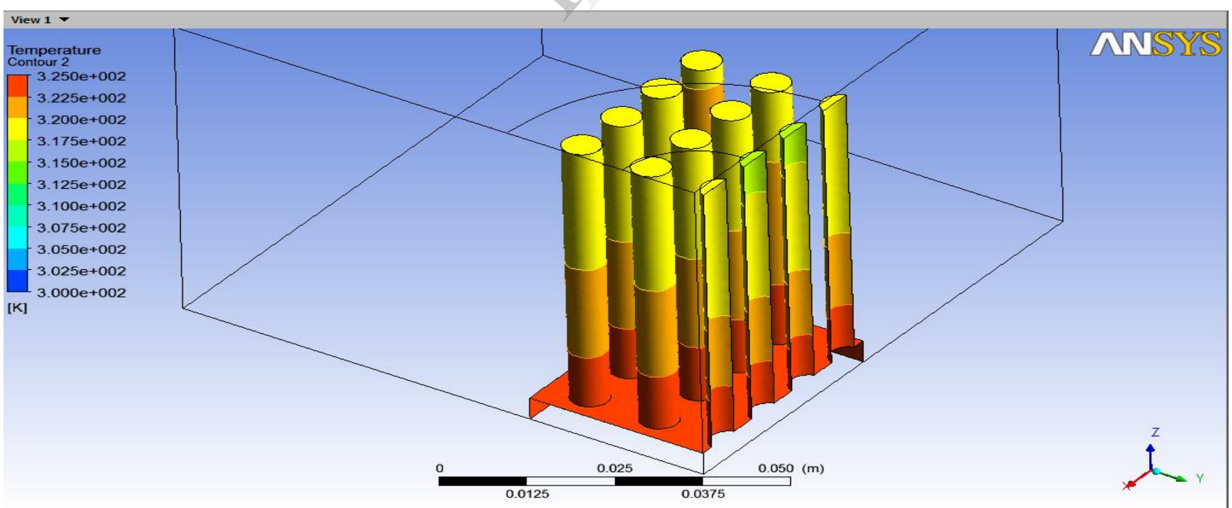


Fig.4 Temperature distribution in Pin fin heat sink with 2.5m/s velocity

T*his fig. shows temperature distribution of air in pin fin. with constant transverse pitch of 12 mm. as fin base has high temperature of 325.K.Air temperature is 300K in surrounding. When air flows, due to convection heat transfer rate increases and temperature of pin fin get* *decreases. In upper portion it is near to 320 K. Because of fin spacing some inner fins have higher temperature upto322.5K*

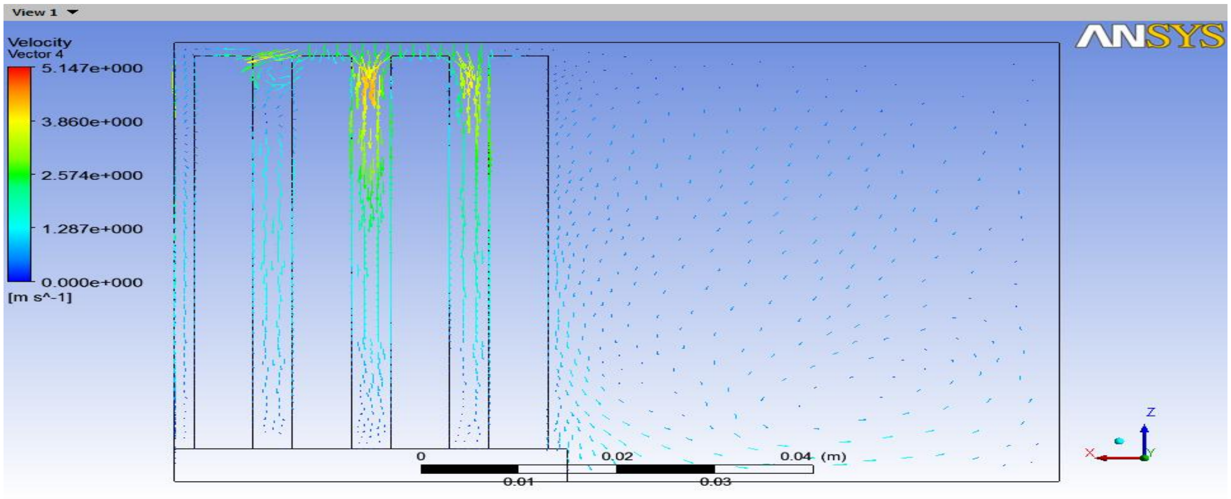


Fig.5Velocity distribution in Pin fin heat sink with 2.5m/s velocity

This fig. shows velocity distribution of air in pin fin of 8mm transverse pitch. As air flows, due to convection heat transfer rate increases and temperature of air gets increases. In upper portion it is near to 5.147m/s.but due to fin spacing when air flows between fins temperature of air increases in some region up to 3.86 m/s.

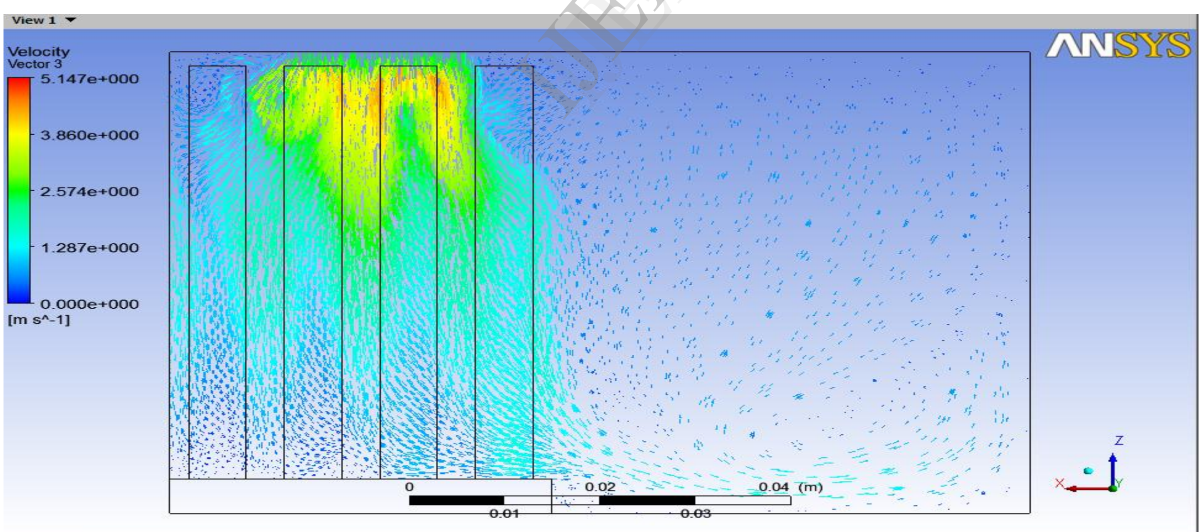


Fig.6Velocity distribution in Pin fin heat sink with 2.5m/s velocity

This fig. shows velocity distribution of air in pin fin of 8mm transverse pitch. As air flows, due to convection heat transfer rate increases and temperature of air gets increases. In upper portion its velocity near to 5.147m/s.but due to fin spacing when air flows between fins temperature of air increases in some region and velocity also increases in this region up to 3.86 m/s.

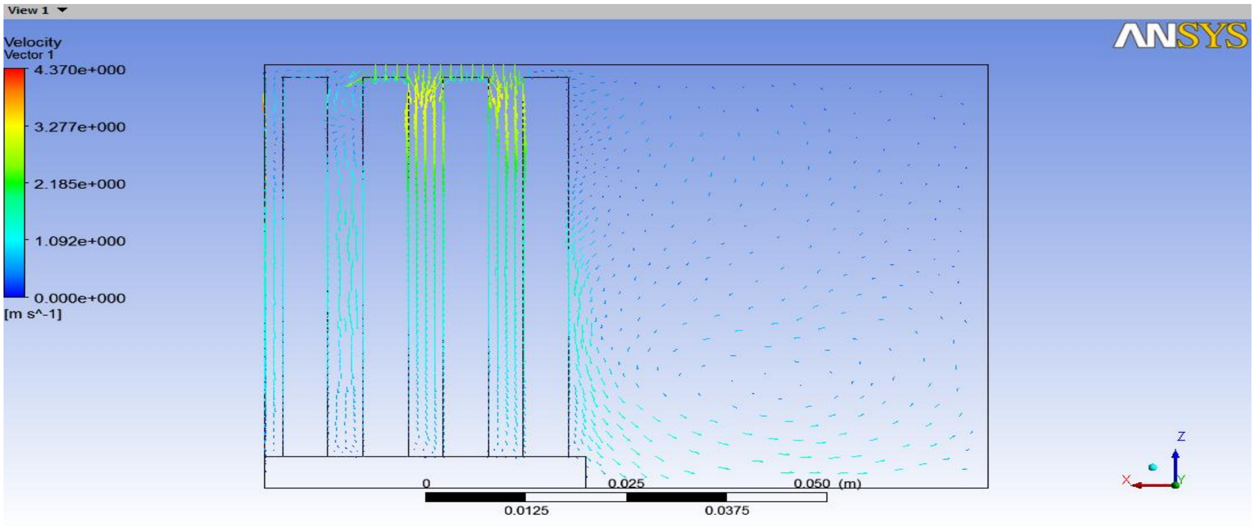


Fig.7 Velocity distribution in Pin fin heat sink with 2.5m/s velocity

This fig. shows velocity distribution of air in pin fin of 12 mm transverse pitch. As air flows initially with 2.5m/s but due to convection heat transfer rate increases and temperature of air gets increases and velocity also increases In upper portion its velocity near to 2.185 m/s. but due to fin spacing when air flows between fins temperature of air increases and velocity also increases in this region up to 3.277 m/s.air flows in fins with velocity of 1.092m/s.

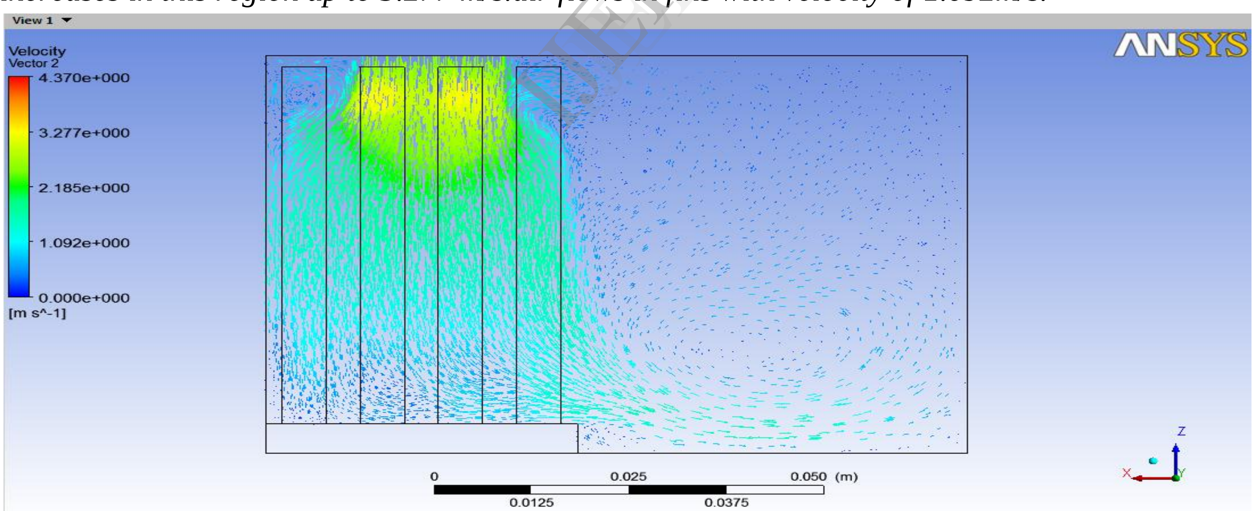


Fig.8 Velocity distribution in Pin fin heat sink with 2.5m/s velocity

This fig. shows velocity distribution of air in pin fin of 12 mm transverse pitch. As air flows initially with 2.5m/s but due to convection heat transfer rate increases and temperature of air gets increases and velocity also increases In upper portion its velocity above 2.185 m/s. but due to fin spacing when air flows between fins, heat transfer rate increases. Temperature of air increases and velocity also increases in this region up to 3.277 m/s.air flows in fins with velocity of 1.092m/s.

CONCLUSION:

Forced convection heat transfer from circular pin fin arrays are investigated by the aid of an academic CFD program, FLUENT 14. The main objective of this study is to determine the thermal performance of pin fins &compare experimental results with CFD results of pin fins for constant fin height of 60 mm, fin diameter 6 mm& constant longitudinal pitch of 10 mmdimensions under forced convection heat transfer. Air velocity is taken as 2.5 m/s. The comparison is done on the basis of, thermal resistanceand temperaturetheconstant power input value of 40 KW.For the experimental setup pin fin surface has some roughness so because of this thermal resistance decreases from .193 **°C/W** to .189 **°C/W**and heat transfer rate increases. But in simulation pin surface consider as smooth surface. Becauseof this smoothness heat transfer rate increases and thermal resistance decreases from 0.475052 **°C/W** to 0.452928 **°C/W.**Finspacing is also using as a tool to compare the results. The results obtained from the analyses shows thatwhen transverse pitch of circular pin fin increases thermal resistance is decreases. As we have the relative relation of simulation and experimental with some considerable deviation.

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