Numerical Investigation of Cylindrical Pin Fin Heat Sink in Natural Convection

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Abstract—**Heat produced by electronic devices and circuitry must be dissipated to improve reliability and prevent premature failure. Heat sinks are the most common thermal management hardware used in electronics. They improve the thermal control of electronic components, assemblies, and modules by enhancing their surface area through the use of pin fins. The objective of this paper is to present an Optimal Heat Sink for efficient cooling of electronic devices. This paper aims at understanding of flow and heat transfer from horizontal base cylindrical pin-fin heat sinks having inline configuration with exposed edges in free convection of air. The study consists of different arrangement of fin height keeping other parameters such as fin base area; fin density and diameter are kept constant which gives maximum heat transfer so that which would obtain an optimal fin type. Here basically three types of fin arrangement are studied numerically, modeled using the ANSYS 14.0 software. A relative contribution of outer and inner fin rows in the sink is assessed from flow and static temperature contour together with the effect of fin location in the array on the heat-transfer rate from an individual fin.**

Index Terms - **Natural convection, Pin fin, flow, heat transfer.**

I. INTRODUCTION

Pin fins are broadly used in thermal management of various applications such as cooling of turbine blades, electronic cooling devices, because of low cost and high reliability. Also rapid increase in the chip speed of laptop and portable electronic devices has led to a large increase in device heat loads. For portables, the lack of sufficient device surface area means that all surfaces must be utilized optimally to achieve cooling. In this paper, a study is performed to understand the limits of free convection for this arrangement and to determine the optimal fin geometry to maximize thermal performance. The optimum thermal design of electronic devices cooled by natural convection depends on an accurate choice of the geometrical configuration and on the heat source position to be able to improve the cooling process. Free convective heat transfer from cylinders having circular and square cross sections has significant practical importance and is relevant to numbers of applications in the cooling of electrical and electronic components such as square pin fin heat sinks, resistors, capacitors. Many researchers had established a number of important results for pin-fin heat sinks in natural convection, concerning sink orientation, additivity of natural convection and radiation and existence of optimum fin population. In particular, they showed that a horizontal-base sink with upward-facing fins outperforms an identical sink with a vertical base and horizontal fins. The ratio of the fin diameter to the Centre-to-Centre spacing was considered, and it was established that the optimum value of this ratio is about 1/3, whereas heat transfer from more densely populated sinks is less effective. The numerical simulation indicated that in such systems an optimum exists, beyond which an increasing number of fins inhibits rather than enhances the heat-transfer rate. In the present study, the effects of fin height on the performance of a horizontal-base pin-fin heat sink with base surface and fin surfaces are exposed to the ambient are studied both experimentally and numerically.

The main problem investigated from literature survey is that in horizontal-base pin-fin heat sinks the outer rows, which are exposed to free flow of ambient air, contribute the major part of the total heat transfer to the surroundings and an individual outer-row fin contributes much more than an inner fin. So for this reason when moving towards the Centre region; the contribution of heat transfer from the fin decreases thus less effective there. The one way of overcoming this effect is removing fins located near the Centre region or trying with a fin of variable height. Thus using this idea various types of heat sink with effect of varying height is studied here.

II. COMPUTATIONAL METHODOLOGY

A. Geometry Creation

Model consists of a horizontal base cylindrical pin-fin heat sink and insulation enclosed in a relatively large box open from top where base plate and pin fins are exposed to free convection of surrounding air. The sink structure is made of Aluminum 6061 with constant base area 100 x 100 mm^2 and 10 mm thick. So the material properties used in the simulation is based on Aluminum 6061. Each pin fin has constant diameter D=4 mm whereas fin height H is varied from outer to inner row. In the present study as discussed before three different pattern of varying pin fin height are tried along with base plate only. The power inputs explored in the present study are 13 W and 20 W. So base plate is taken as a heat generating core where plate side walls and bottom surface are insulated. The base and fin surface are exposed to ambient air. The outer vertical extensions of the domain also insulated with a Perspex material. The sink is enclosed in an expanded polystyrene [EPS] insulation so that the upper surface of the base is mounted flush with the upper surface of the insulation, which does not touch the fins, thus allowing flow of air from the edges to the fin array. The insulation thickness is 145 mm on each side of the sink, and about 135 m m under the sink. The outer box dimensions are about 390 x 390 x 395 mm^3 with negligible wall thickness.

As said before three different fin types are created in ANSYS pre- processor stage. A typical one (Type A) is shown in fig. 1. In the case of Type A pin fin height is reduced from outer to inner row. It is essential to note that the "rows" are defined in the following manner: the "first row" denotes all the fins located at the outside perimeter of the sink, the "second row" means the next row inside, and so on. For the case of Type A, the height of sink reduces from first to fifth row as 40, 35, 30, 25, 20 mm respectively. Then for Type B fin height is increased from outer to inner row as 20, 25, 30, 35, 40 mm respectively. A uniform fin height of 40 mm is provided for Type C. A model created in CATIA V5 as shown in Fig. 2 gives an idea about sink structure where fin height is varied from outer to inner row.

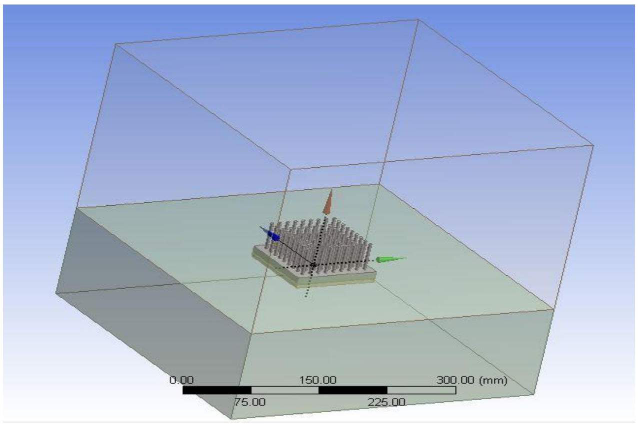

Figure 1 Full geometry showing sinks structure and insulation.

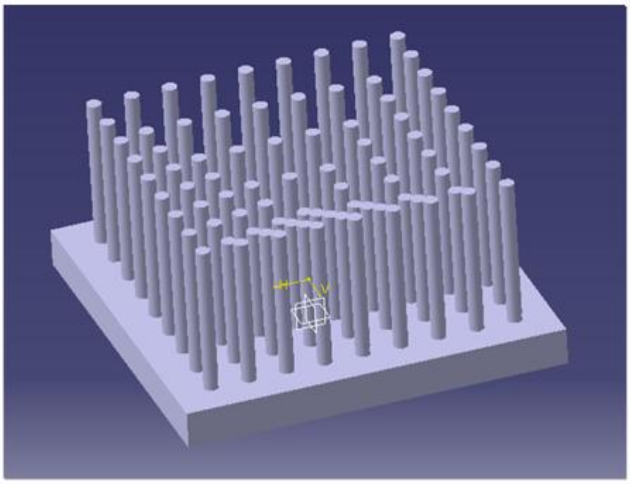

Figure 2 Model created in CATIA V5 showing sink structure.

B. Boundary Conditions

Zero heat flux boundary condition is imposed on all insulating surfaces. Pressure outlet condition is used for the open surface above the box. For the momentum equation (air) no-penetration and no-slip are assumed at all the solid boundaries exposed sink, insulation surfaces and vertical walls. Due to the plane of symmetry only quarter portion of the domain and heat sink is selected for computational study in order to reduce computation time as shown in Fig. 3. In simulations time independent basic conservation for continuity, momentum and energy equations are solved numerically in FLUENT 14.0.

In the numerical simulation pressure-based solver is used. Pressure-velocity coupling is applied. PRESTO discretization is used for the pressure; Second-order upwind discretization is used for the momentum and energy. The software enables the calculation of the overall rate of change of the internal energy of the air, as well as the calculation of the overall heat input or output at any boundary. Thus, these results served to ensure the overall energy balance of the system. The convergence criteria is set for continuity and momentum are 10^{-4} and for energy 10^{-6}.

A uniform fin diameter is provided for entire row. The array selected for study consists of 9 x 9 arrays. So a total of 81 fins with Centre to Centre distance (pitch) equal to S=10 mm. Thus fin density and fin spacing are kept constant and pin fin height is varied for analysis of current study.

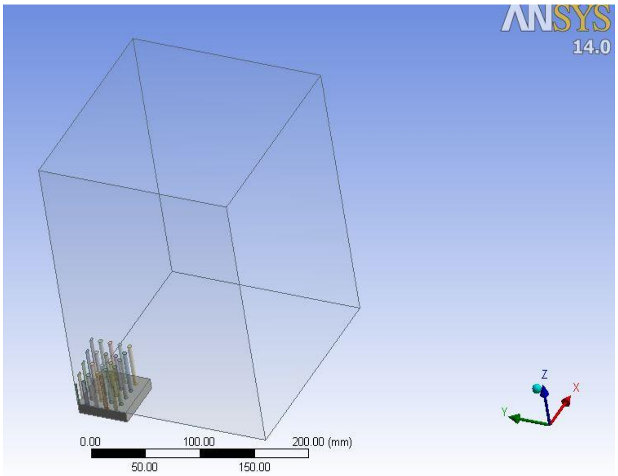

Figure 3 The final geometry created in ANSYS 14.0 pre-processor stage

C. Governing Equations

The analysis of natural convection heat transfer is in general a comparatively complicated process. It is required to determine the velocity, temperature and pressure fields simultaneously. The governing equations for three-dimensional, steady state, laminar incompressible buoyancy-induced flows with single phase and constant fluid properties which are used in computational study are;

Continuity,

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

Momentum Equation

$$\frac{\partial}{\partial x_i}(\rho u_i u_j) = \rho g_i - \frac{\partial p}{\partial x_i} + \frac{\partial \tau_{ij}}{\partial x_j} \qquad (2)$$

Energy Equation,

$$\frac{\partial}{\partial x_i}(\rho u_i h_e) = \frac{\partial}{\partial x_i}\left(k \frac{\partial T}{\partial x_i}\right) \qquad (3)$$

Where ρ is the density of air and *ui gi* are velocity component and acceleration due to gravity in *i* direction respectively. P is the static pressure, k is the thermal conductivity, h_e is the enthalpy and T is the temperature. Since the flow is buoyancy-driven the static pressure in the momentum equation is redefined by the following relation for the vertical direction (say z direction): P' $=\rho_0$gz +P, where ρ_0 is the reference density taken at ambient temperature and atmospheric pressure. The Boussinesq approximation is not used instead air is assumed as incompressible-ideal gas. It should be noted that the three-dimensional numerical calculations are performed for the system as a whole. This means that the velocity fields and the temperature distributions in the air, the temperature distributions in the heat sink etc.

D. Grid Used

The grid size used for the simulation is not uniform. Tetrahedral mesh with variable size is used. It is important that finer mesh is provided near the solid boundaries, especially at the base and fins. The total numbers of elements in current study cross a value of 1.7 x 10^6. A typical mesh generated for Type A heat sink is shown below.

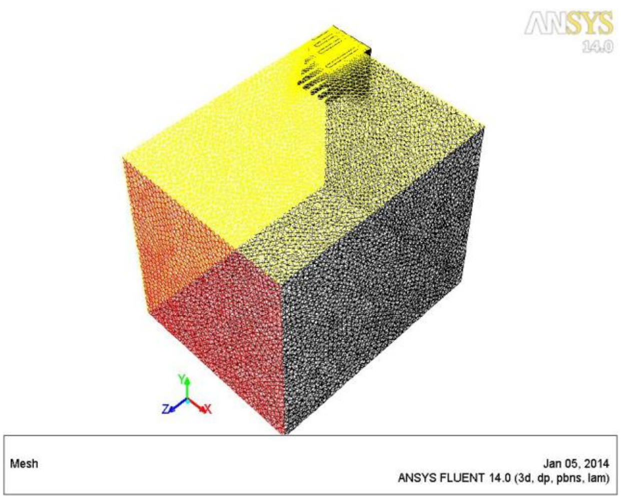

Figure 4 Mesh generated for a Type A heat sink.

III. RESULTS AND DISCUSSION

Numerical study on horizontal base vertically oriented cylindrical pin fin array in free convection is done for a constant fin density, base area and spacing between fins. The numerical study on three different sink types were done at a heat inputs of 13 W and 20 W.

A. Temperature Contours

The temperature distribution in the central plane of the heat sink (pin fin array) is shown in figure 5 to figure 7 for three types of heat sinks for a heat input of 20 W. It is evident from the figures that the central region of the fin array has the maximum temperature. This is due to the fact that the outer rows of pin fins impede the flow of fresh air over the central part of the array, which leads to lower heat transfer from the central region, leading to higher temperatures. The outer rows of the fin array are at considerably lower temperatures, because it is in intimate contact with the surrounding cooler air.

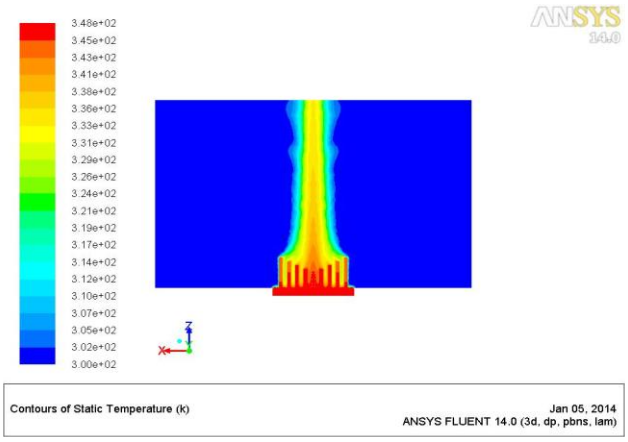

Figure 5 Temperature distribution in sink (Type A) and air for 20 W heat input.

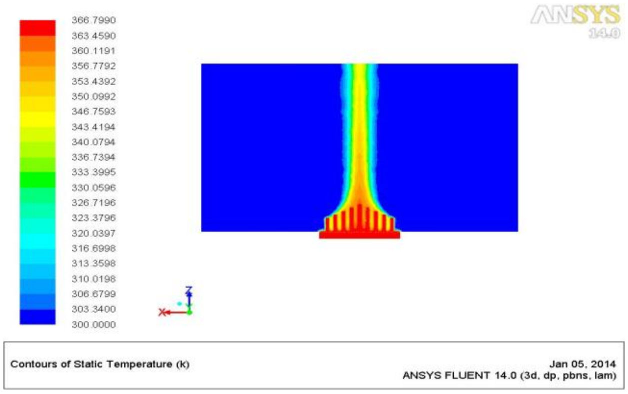

Figure 6 Temperature distributions in sink (Type B) and air for 20 W heat input

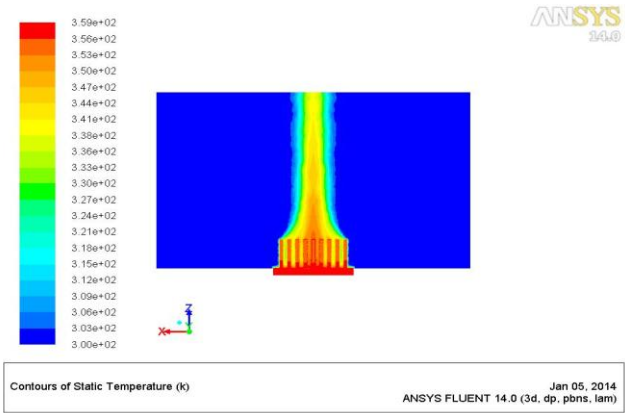

Figure 7 Temperature distributions in sink (Type C) and air for 20 W heat input

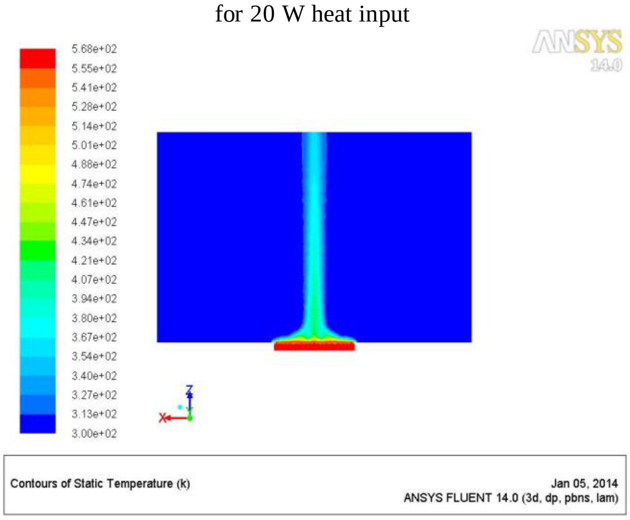

Figure 8 Temperature distributions in a flat plate and air for 20 W heat input.

The simulation indicates that the air enters the sink at a lower velocity and velocity is again reduced at the sink Centre. This, together with the fact that the air temperature increases as it flows across the sink, makes the internal fins highly inefficient in those configurations in which the outside perimeter of the fins is exposed to the ambient.

As evident from the figures 5 to 7, the temperature difference, T_W-T_∞ (T_W -average sink temperature and T_∞ -ambient temperature) is minimum for the fin array of type A which makes it the best performing fin out of the three types considered in the current study. Due to the high thermal conductivity of the heat sink material and relatively high convective resistance, the surface of the heat sink will be almost at uniform temperature. In the outer region cool air is continuously in intimate contact with the surface of the heat sink, so the heat fluxes will be high in that region.

Type B is the least performing array among the three types. Sink temperature is maximum at the base. Also for a flat plate it was found that the sink temperatures reaching a value of 568 K because of low heat transfer from the exposed sink base. For a flat plate heat transfer area is minimum and convective heat transfer is minimum thus following the relation heat input, $Q = h\ A\ \Delta T$ where h is the heat transfer coefficient, A is the exposed sink base area and ΔT the temperature difference.

Heat transfer from a horizontal surface is complicated by the fact that the heat transfer may be with a fluid either above or below a horizontal plane and the surface may be either hotter or colder than the fluid. These factors are important because heated fluid will rise and cooled fluid will sink due to the fluid temperature change.

B. *Maximum Temperature*

From Figure 8 it can be understood that the maximum temperature recorded for a type A heat sink is the minimum among the three types. When height of the fin is gradually increased from outer to inner row (type B) the maximum sink temperature is the highest. The maximum sink temperature for a uniform fin height lies in between type A and type B.

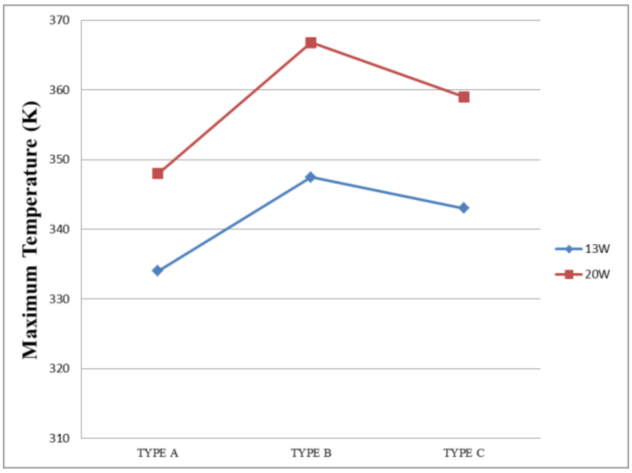

Figure 8 Graph showing maximum sink temperature for different sink types for different heat inputs of 13W and 20W.

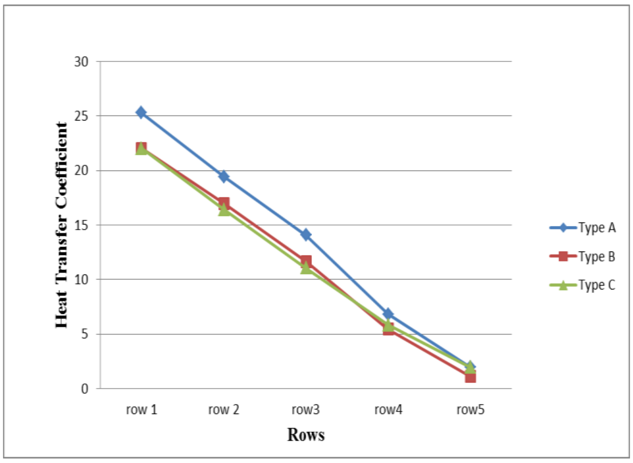

Figure 9 Average heat transfer coefficient in each for 20 W heat input.

C. *Heat Transfer Coefficient*

For each heat input, the average heat transfer coefficient decreases. For low and high heat inputs the Type A outperforms the other types. In all the three cases the fins in the outer row performs better for both heat inputs.

Fin array of type B, where fin height is increases from outer to inner region, starting from 20 mm height to 40 mm the heat transfer in the 5th row is slightly high compared to Type A because fin height in the fifth row ,in Type B is more compared to Type A. For Type C fin height is same for entire row. Fig. 9 shows the averaged heat transfer coefficient in each rows (outer to inner rows) for a 20 W heat input. In the case of 13 W heat input the trend is similar and that one is not shown.

D. *Flow Structure*

The figures 10 to 12 shows the flow structure for 20 W heat input. The flow at the central region of fin array is shown for all three types of fin arrays where flow enters in a horizontal direction towards the outer fin array and gets heated up from there. But flow direction changes to vertical when flow reaches exactly at the Centre and moves with a greater velocity of about 0.8 m/s. So flow is laminar everywhere in the flow domain. Also a circulation of flow can be seen on the top of outer region of the fin. This is because heated air is replaced with surrounding cooler air. As seen in the earlier temperature contours, the inner rows especially fourth and fifth row of fin has more surface temperature than outer fin because the density of air decreases due to heating effect and amount of air required for continuous cooling is less due to the obstruction made by outer rows. This is because flow velocity decreases due to friction between air and solid surface. The cool air flows into the sink, both with and without the internal fins, mostly from the sides, and the flow patterns in all cases are similar and resemble.

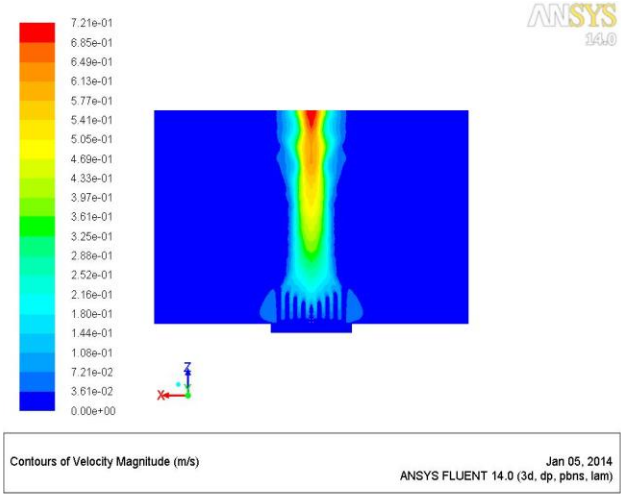

Figure 10 Air flow pattern near sink (Type A) for 20 W heat input.

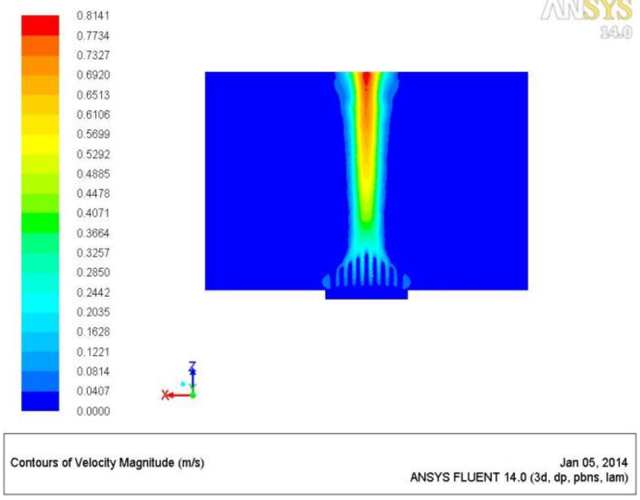

Figure 11 Air flow pattern near sink (Type B) for 20 W heat input

Numerical study indicates that Type A heat sink having cylindrical pin fins is good at both flow and heat transfer study with a considerable reduction in weight compared to convectional rectangular pin fins used.

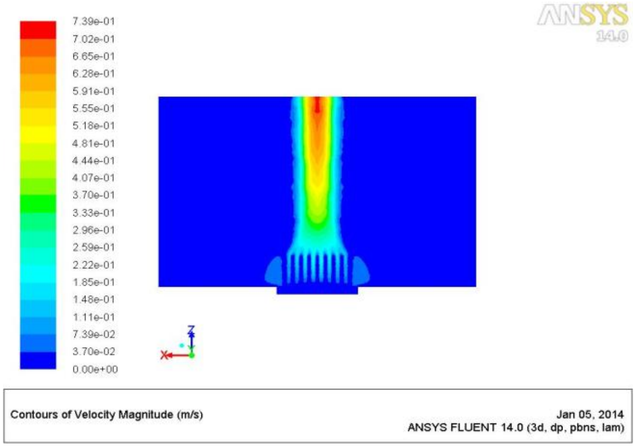

Figure 12 Air flow pattern near sink (Type C) for 20 W heat input.

IV. CONCLUSIONS

Numerical investigation on horizontal base cylindrical pin fin heat sink in natural convection is done. The useful information of sink performance is obtained from temperature and flow analysis. If fins are made of cylindrical shaped ones the resistance to the air flow reduces as compared to rectangular fins that are common.

The Type A heat sink is better performing among Type B and Type C. The typical air flow pattern is chimney like. That is flow enters horizontally and moves vertically with a greater velocity compared to inlet stream. The heat transfer from the outer row of fin is better performing one especially for Type A and Type B. Also the heat transfer from individual pin fin in outer row is high. Since Type A is better performing one the material and sink weight is reduced considerably and in design point of view but the problem is when we go for manufacturing of such sink structure.

V. Acknowledgment

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