A Numerical Study on Dynamics of Heat transfer in Electronic Cooling

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Abstract

Natural/Free convection, a very common mode of heat transfer which has immense practical applications, attracted the minds of scientists and engineers for the great many years. Here, the free convection taking place from the heat sink of an electronic device is elucidated numerically using a commercially available software-FLUENT. A 2D incompressible simulation is performed for the laminar thermal convection. The mesh used for the simulation purpose is generated by using software called GAMBIT. It has already been observed experimentally that the thermal plumes have got premiere responsibility for the heat transfer mechanism happening in free convection. We support our numerical investigations by the previous experimental works. The flow is constrained well within the range of laminar flow in terms of Rayleigh number. In the present study, we have studied the effect of different parameters such as aspect ratio, separation between the heat sinks, Rayleigh number etc. on the heat transfer from the electronic chip and also understood how plume generation aids heat transfer in electronic chip cooling.

Keywords—Electronic cooling; Natural convection; Thermal plumes; Heat transfer enhancement.

I. INTRODUCTION

Since from the middle of 1940, the first electronic computer and devices were discovered, the technology has come a long way. Faster and smaller computers have led to the development of denser and smaller circuit technologies which further has led to increased heat fluxes generating at the chip and the package level. Over the years, significant advances have been made in the application of air cooling techniques to manage increased heat fluxes. Air cooling continues to be the most widely used method of cooling electronic components because this method is easy to incorporate and is cheaply available. Although significant heat fluxes can be accommodated with the use of liquid cooling, its use is still limited in most extreme cases where there is no choice available.

The low thermal conductivity of commonly used organic materials intensifies the impact of this high heat flux by causing large temperature gradients between components and their substrate. Finned, air-cooled heat sinks and liquid cooling are alternative designs for heat removal, but the major drawbacks of these designs are the increases in weight, cost, and volume. Natural convection provides a means to facilitate and enhance heat and mass transfer for microelectronic devices. Natural convection is also an area of interest for enhanced heat and mass transfer for recently developed biochemical analysis systems and micro-fuel cell designs.

Numerous studies have been performed on horizontal fluid layers heated from below. A growing body of work also exists in the area of convection due to discrete heat sources because of its fundamental interest and relevance to applications in electronics. Incropera et al.[1] performed experiments to determine the heat transfer due to conduction and forced convection from a four-row array of heat sources flush-mounted on one wall of a horizontal channel filled with water. Heindel et al.[2,3] studied the natural convection from an array of discrete heat sources in a cavity filled with water and a dielectric fluid. They also investigated the natural-convection heat transfer for an array of finned, discrete heat sources in a cavity filled with a dielectric fluid. Ortega and Lall [4] performed experiments to measure the heat transfer coefficient on the surface of a square flush-mounted heat source at the center of a plate in a small horizontal enclosure. Deng et al.[5] presented a two-dimensional numerical investigation of natural convection from two discrete flush-mounted heat sources in a horizontal enclosure with insulating side walls at steady state to investigate the interaction between sources. Tou and Zhang[6] presented a three-dimensional numerical model to investigate the heat transport in a liquid-filled vertical rectangular enclosure with a 3*3 array of discrete flush-mounted heaters along one vertical wall. The opposite wall acted as a uniform cold surface, and all other walls were insulting. Bae and Hyun[7] studied two-dimensional laminar natural-convective air cooling in a vertical rectangular enclosure with three discrete flush mounted heaters on one side of the wall. The thermal condition of the lowest-elevation heater alternated between ‘‘on’’ and ‘‘off,’’ and the resulting effect on transient heat transfer for the other sources was studied. Da Silva et al.[8] Investigated the optimum distribution of heat sources cooled by laminar natural convection for a small number of heat sources mounted on the side wall of an enclosure and for a large number of heat sources mounted on a vertical wall facing a fluid reservoir. Da Silva et al. [9] also investigated the

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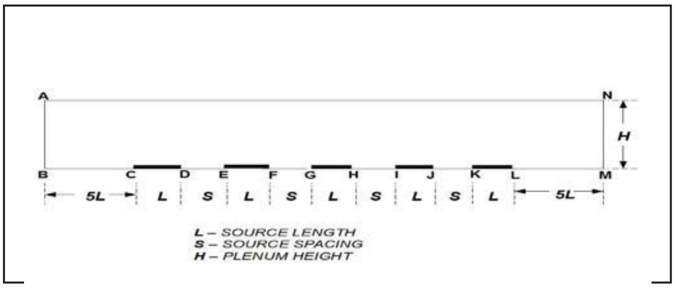
optimal distribution and sizes of three discrete heat sources in a vertical open channel cooled by natural convection. Tso et al. [10] presented experimental and numerical results for laminar natural-convection cooling of water in a rectangular cavity with a 3*3 array of heaters on one wall at various angles of inclination. Papanicolaou and Gopalakrishna [11] presented a two-dimensional computational investigation of natural convection in a shallow horizontal air layer driven by a single flush mounted, discrete, constant heat flux source. They investigated the parameters governing the transition from the conduction-dominated regime to a convection- dominated regime. The geometric parameters studied in their work are the width-to-height aspect ratio of the air layer to the uniformly heated source size. With a uniform heat source, a discrete transition region was observed, whereas with discrete heating, the transition was continuous. For each source size, an optimum aspect ratio for heat transfer was found. In the presence of three discrete heat sources, they found that the transition from conduction to convection was significantly delayed in the presence of adjacent sources compared to the single source; however, the rate of increase of Nusselt number with increasing Rayleigh number was higher in the case of multiple heat sources.

The design of more efficient passive cooling for high-density packaging of electronic devices requires a better understanding of the parameters governing natural-convective heat transfer in the basic arrangement of a horizontal enclosure with many discrete heat sources. This work provides new insight on the physical parameters, such as source length and spacing, which control the onset and generation of sustained natural convection in a horizontal air layer with a large number of heat-generating components. We determine the effects of source spacing and source lengths on heat transfer rates and flow regimes.



The physical problem, analyzed in the present work, is the heat transfer generated by natural convection in a plenum having a number of chips with specific S ,H and L values. The source height is neglected. In some cases, the computational domain size requirements change with the regime, as will be discussed later. In this work, the maximum temperature difference is assumed to be small enough to justify the use of the Boussinesq approximation, the buoyancy induced flow is assumed to be 2D, laminar, and the fluid studied is incompressible with constant thermal and flow properties. Flush-mounted heat sources of equal length, L, have prescribed isothermal conditions. The heat sources are separated by length S, and the height of the computational domain is H. The top and bottom boundaries are specified with no-slip wall and isothermal conditions. A periodic boundary condition is assigned to the left and right boundaries.

This study addresses the effect of H/L ratio, S/L ratio and Rayleigh number towards the Nusselt number and the optimum condition for heat transfer is found out.

II. Numerical Method

A 2D incompressible simulation is performed for the laminar natural convection. The physical domain, as shown in figure 1 is created and meshed with the help of a commercial package called GAMBIT. The domain contains 1000 quadrilateral cells with 1200 nodal points. The governing partial differential equations such as continuity, momentum and energy equations are solved in the discretised domain by using FLUENT, a finite volume solver. The segregated solver solves conservation governing equations independently. The second order upwind differencing scheme was used for momentum and energy equations. The discretization scheme used for pressure was PRESTO. The pressure–velocity coupling is ensured using the SIMPLE algorithm.

III. Results and Discussions

In this study, we found that the main factors influencing the heat transfer behavior for the configuration as in figure 1 are

- The Rayleigh number
- The H/L ratio
- The S/L ratio

We tried to establish the reason behind the changes in heat transfer behavior with these parameters and also put efforts in quantifying it with the help of average Nusselt number, the non dimensional heat transfer coefficient. The isotherms and the stream lines are plotted for understanding the flow physics and the heat transfer characteristics in this particular problem.

A. *Effect of Rayleigh Number on the chip cooling*

Raleigh number (Ra) represents the buoyancy effects inside the enclosure over the viscous effects. In this section, the effect of Rayleigh number on the heat transfer behavior is studied in the range of $[10^{-1},10^{5}]$ for H/L ratio=1, 2 and S/L ratio=1. It is seen that the Rayleigh number has a positive effect on the overall heat transferred from the hot electronic chips inside the enclosure. As the Ra increases, heat transfer increases, for the range of Ra considered in the present work. This can be confirmed from figure 2 where the Nusselt number (Nu) is plotted against Ra. It is reported in the previous literature[11] that the mode of heat transfer, below certain critical value of Ra is conduction dominated.

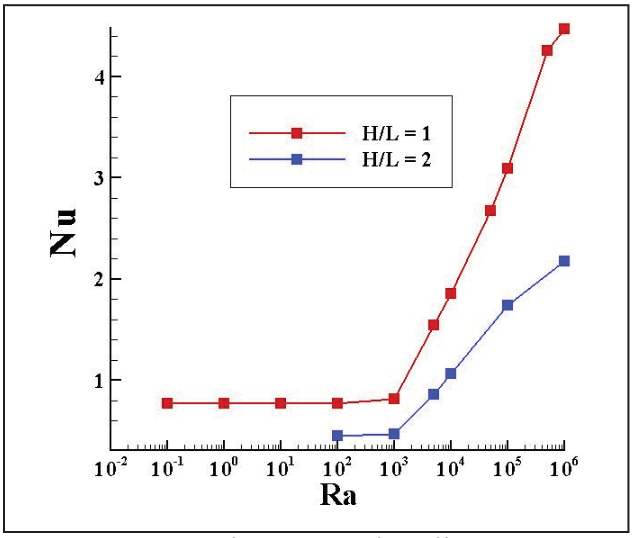


Fig. 2. Variation of Nu with Ra for different H/L ratio

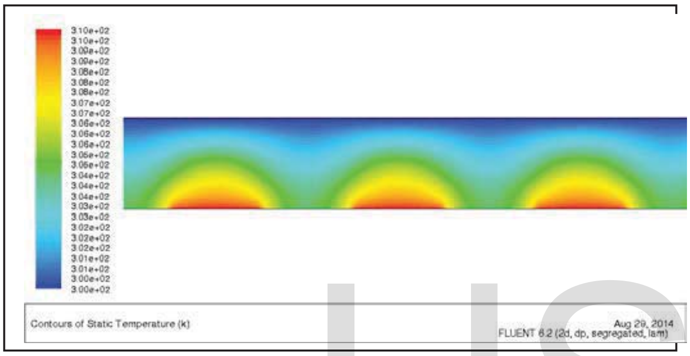


Fig. 3. Isotherm plot for Ra=10^3

In figure 2, a rapid increase in Nu can be seen over Ra=10^3, which means an enhancement in heat transfer above this critical value of Ra. We know that the convection mode of heat transfer plays a major role in heat transfer, over conduction. In the present analysis, the flow becomes convection dominated over the conduction, for Ra greater than 10^3 and is attributed to the sudden jump of Nu plot about Ra=10^3 as in figure 2.

In order to establish the physics more clearly, we have plotted the isotherms above the hot chips for Ra varying from 10^3 to 10^5 for H/L=1 and S/L=1 as shown in figures 3-6. It should be noted that we have taken isotherms of the same value in all the four cases for comparison. For Ra=1000, there were no plumes observed from the hot chips. Plumes are essential coherent structures formed due to thermal boundary layer instability, which helps to convect heat from hot surfaces. In the present analysis, thermal plumes are observed for Ra=10^4 and 10^5 as shown in figures 5 and 6.The hot chips generates a density stratification inside the enclosure due to the temperature gradient present and buoyancy will accelerate the fluid in the upward direction and is the reason for the origin of thermal plumes. The presence of plume increases the upward velocity of the fluid due to buoyancy inside the enclosure in an otherwise, almost stagnant condition. This is the reason for the increased convection current for Ra greater than 10^3. Also plumes generates more number of Rayleigh Benard (RB) convection cells as shown in figures 6, where the stream lines are plotted inside the enclosure. The number of streamlines taken in both the figure 5 and 6 is the same for comparison. However, in figure 5 (Ra=10^3), the number of convection cells are lesser than in figure 6 (Ra=10^5). It should be noted that the plumes are observed above Ra=10^3.

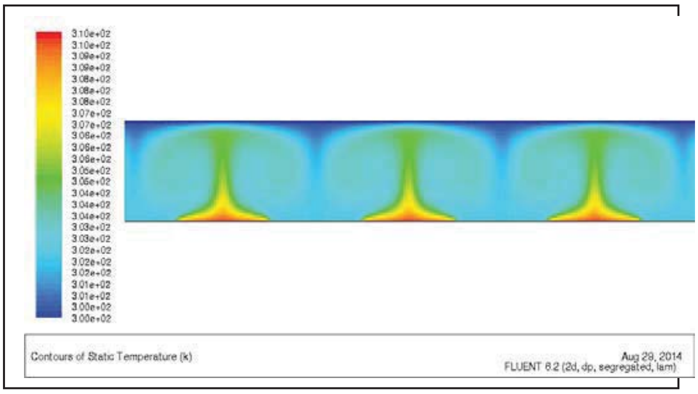


Fig.4. Isotherm plot for Ra=10^5

In short, the increase in Ra increases the number of plumes and thereby increases the number of RB convection cells which is attributed to the increase in heat transfer as Ra gets increased.

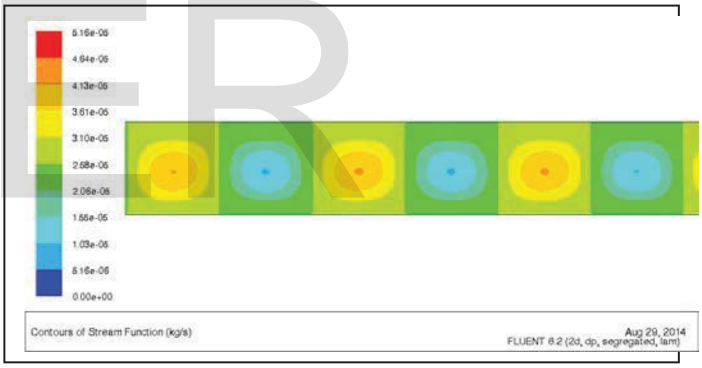


Fig. 5. Streamline plot for Ra=10^3

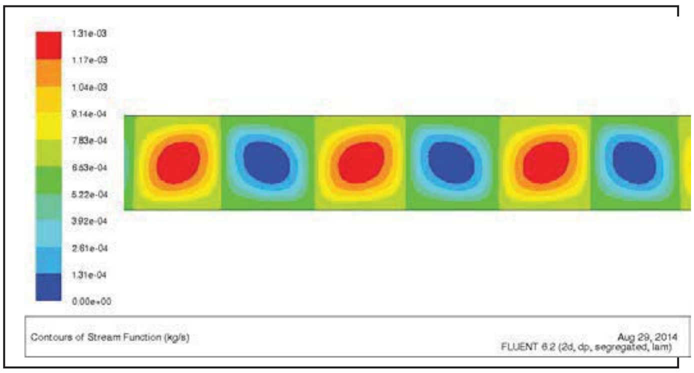


Fig. 6. Streamline plot for Ra=10^5

B. *Effect of H/L ratio*

In this section, the effect of H/L ratio on the heat transfer characteristics is explained. The isotherms (figures 11-14) and streamlines (figures 15-18) are plotted for Ra=10^4 and S/L ratio=1 and different H/L ratios 0.125, 0.25, 1 and 4 respectively. It is explained in the previous section that the number of plumes characterizes the number of RB cells and hence the heat transferred from the chips. An increase in H/L ratio decreases the number of plumes per chip inside the enclosure (figures7-10) and the number of RB cells as shown in figures 11-14. The reduction in RB cells reduces the convection heat transfer. Hence the Nu decreases asymptotically to a constant value as H/L ratio increases as shown in figure 15, which indicates that above this critical H/L ratio, heat transfer is found fairly constant and is independent of H/L ratio for that value of Ra considered.

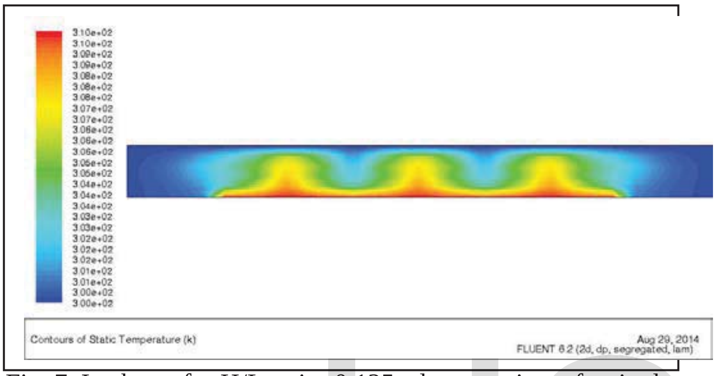


Fig. 7. Isotherm for H/L ratio=0.125, close up view of a single chip

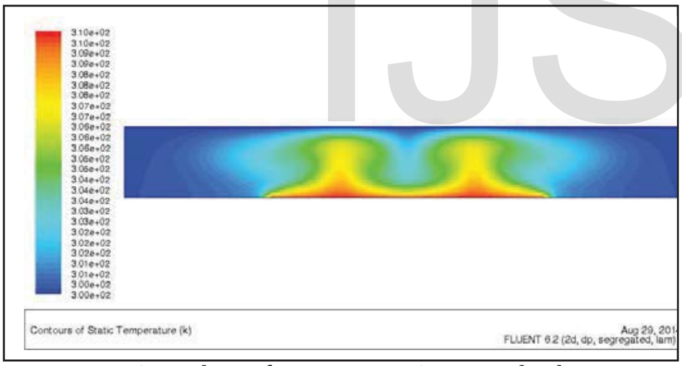


Fig. 8. Isotherm for H/L ratio=0.25, single chip

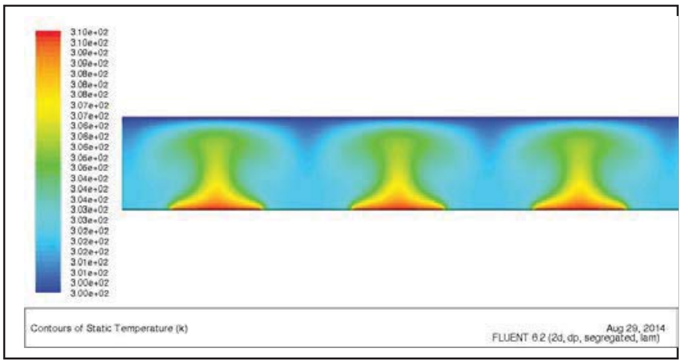


Fig. 9. Isotherm plot for H/L ratio=1, 3 chips

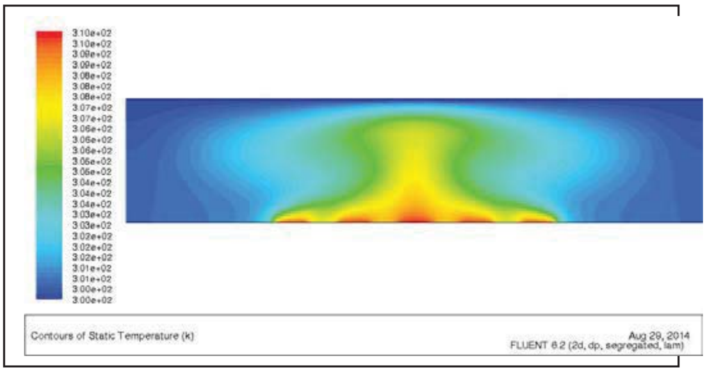


Fig. 10. Isotherm for H/L ratio=4, 5chips

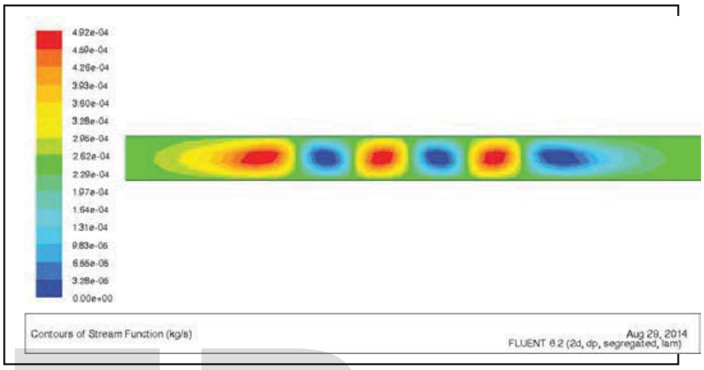


Fig. 11. Streamline plot for H/L ratio=0.125, single chip

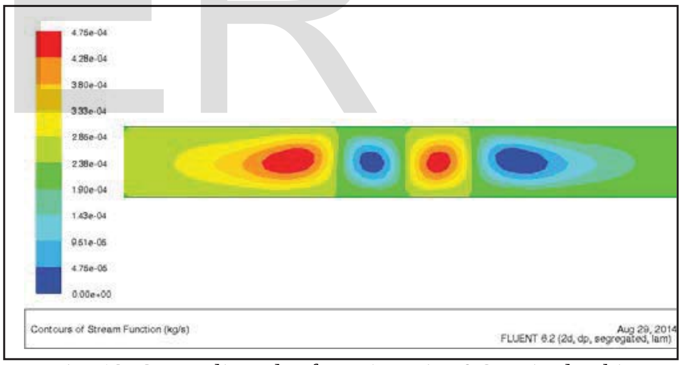


Fig. 12. Streamline plot for H/L ratio=0.25, single chip

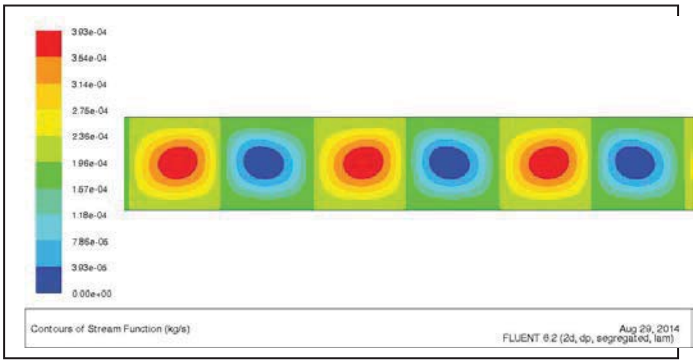


Fig. 13. Streamline plot for H/L ratio=1, 3 chips

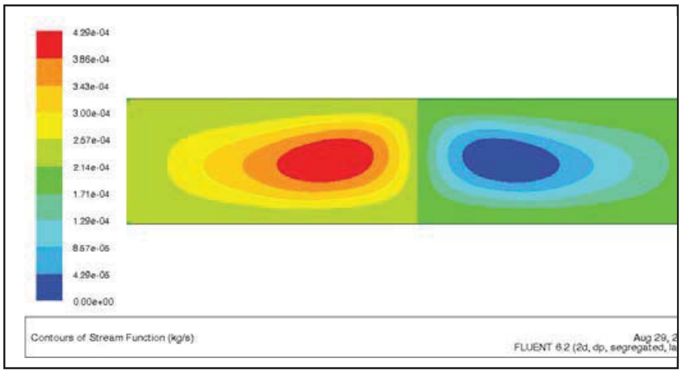


Fig. 14. Streamline plot for H/L ratio=4, 5 chips

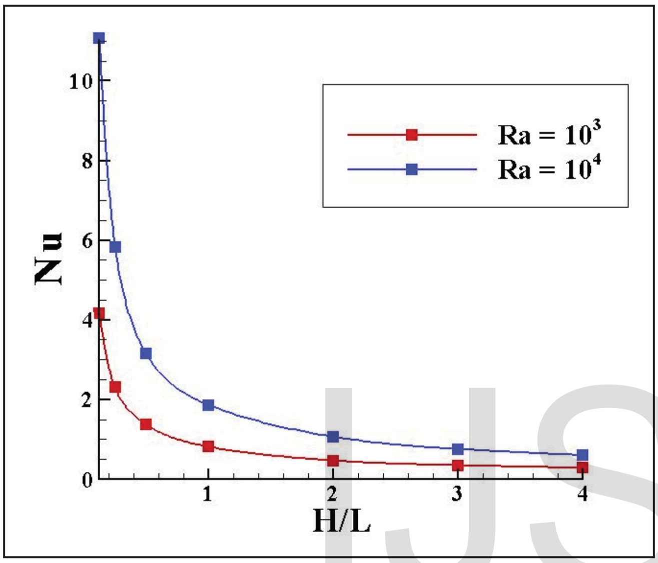


Fig. 15. Variation of Nu with H/L for different Ra values

C. Effect of S/L Ratio

We extended the same analysis to the effect of varying source spacing. Figures 16 and 17 show the isotherms plotted for H/L ratio 1 and for two Ra = 10^4. It can be seen that the plumes from each sources are getting stronger as S/L ratio is increased. The RB cell also becomes stronger with the increase in S/L ratio as seen in figures 18 and 19. Hence the heat transfer increases with the increase in S/L ratio asymptotically to a constant value as seen from figure 20 and 21.

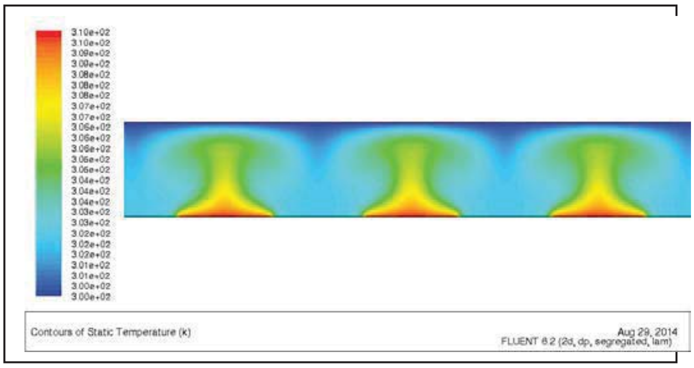


Fig. 16. Isotherm for S/L ratio=1

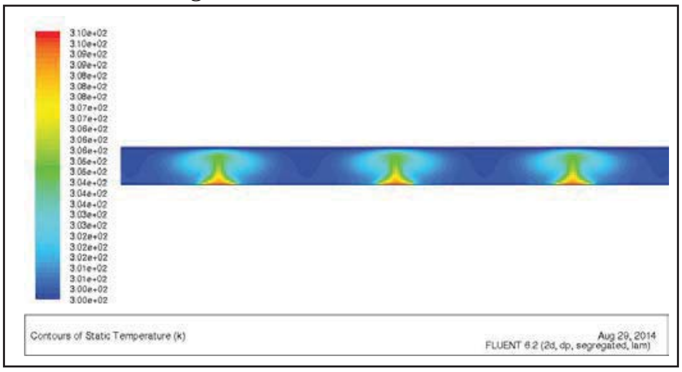


Fig. 17. Isotherm for S/L ratio=4

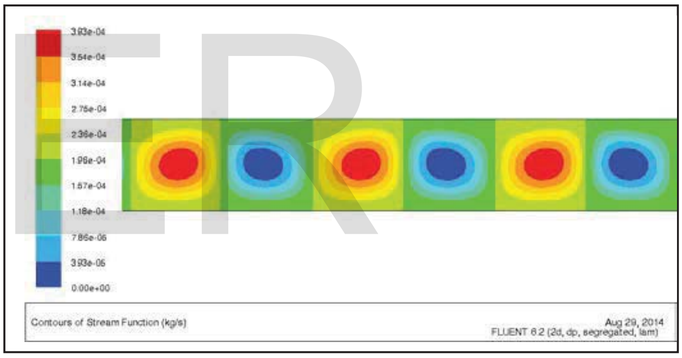


Fig. 18. Streamline plot for S/L ratio=1

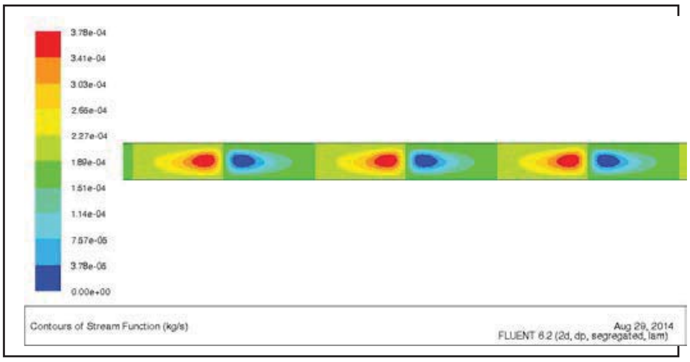


Fig. 19. Streamline plot for S/L ratio=4

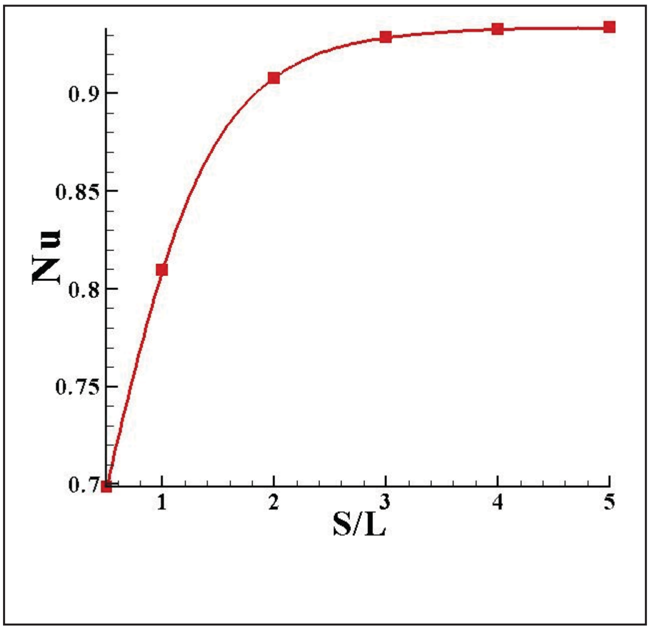


Fig. 20. Variation of nu with S/L ratios for Ra=1000

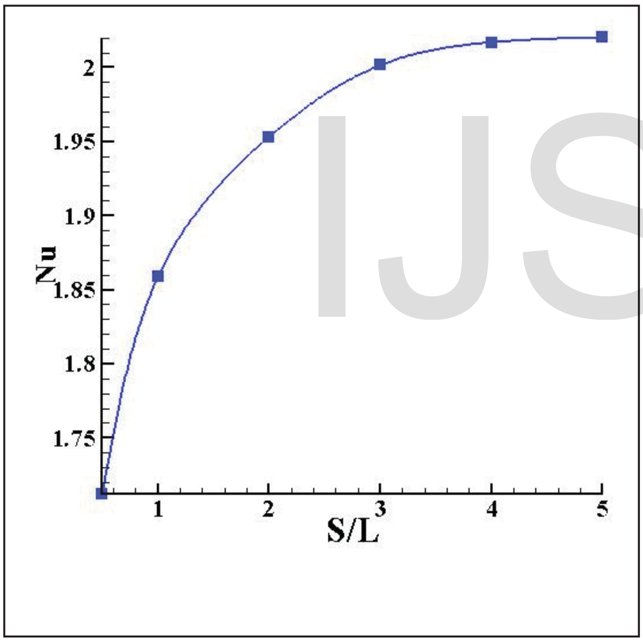


Fig. 21. Variation of nu with S/L ratio for Ra=10000

Nusselt number increases steeply up to S/L ratio equals 3 and after that the nusselt number almost a constant for both the graphs. This is because after a particular spacing length, further increase in spacing will not affect the heat transfer. Isotherms and streamline plots for S/L = 2 and 4 are almost similar. This indicates that optimum spacing will decrease the size of the column containing the sources.

IV. Conclusions

A numerical study is conducted to study the dynamics of heat transfer in electronic cooling using natural convection for understanding the effects of some critical parameters such as Rayleigh number, H/L ratio and S/L ratio. For higher values of the Rayleigh numbers ($Ra>10^3$), convection dominates over conduction and a rapid jump in Nu plot was seen. The average Nusselt number increases with increase in Rayleigh number. We found that the plumes are essential coherent structures which increase the convection heat transfer. Each plume will introduce couple of RB cells which acts as catalyst for heat transfer. We studied the effects of heat transfer with the help of isotherms and streamline plots. Isotherms give the structure of plumes generated and stream lines will give the RB cells. An increase in either H/L ratio or S/L ratio increases the heat transfer to an optimum value asymptotically. Above this value the Nusselt number will not respond to an increase in either H/L or S/L ratio.

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