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# Numerical investigation of natural and mixed convection heat transfer on optimal distribution of discrete heat sources mounted on a substrate

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**Abstract:** The paper deals with the numerical investigation of natural and mixed convection heat transfer on optimal distribution of five non-identical protruding discrete heat sources (Aluminium) mounted on a substrate (Bakelite) board. The heat sources are subjected to a uniform heat flux of  $2000 \text{ W/m}^2$ . The temperature of heat sources along with the effect of thermal interaction between them is predicted by carrying out numerical simulations using ANSYS Icepak, and the results are validated with the existing experimental findings. The results suggest that mixed convection is a better method for cooling of discrete heat source modules. Also, the temperature of heat sources is a strong function of their shape, size, and positioning on the substrate. Effect of radiation is studied by painting the surface of heat sources by black paint. The results conclude that, under natural convection heat transfer, the temperature of heat sources drops by 6-13% from polished to black painted surface, while mixed convection results in the drop by 3-15%. The numerical predictions are in strong agreement with experimental results.

## 1. Introduction

There is a stiff challenge to deal with the increasing heat flux dissipation from electronic components. There are a number of cooling methods widely used in electronic industries, which are broadly categorized into active cooling and passive cooling. Mechanically assisted cooling systems provide active cooling since they require energy. Passive cooling is also equally important, especially in module level thermal management, and for low heat flux removal rates. Conventional cooling methods cannot deal with the heat flux demand rate. Hence, innovative cooling techniques need to be implemented to enhance the heat removal rate in order to maintain their safe operating temperature. The heat flux must be of the order of  $3 \times 10^6 \text{ W/m}^2$  to maintain the temperature below  $85^\circ\text{C}$  for micro and power-electronic industries. Air continues to be the most widely used coolant in electronic systems due to its low-cost, ease of availability, less maintenance, non-contamination, and most importantly, does not add vibrations, noise, and humidity to the system.

## 2. Literature review

A detailed review has been carried out on the numerical investigation of heat sources. Chen and Liu. [1] conducted experiments to determine the optimum distribution of nine identical protruding heat sources, under forced convection, and suggested that conventional equally spaced heat source are not the best technique and a better heat transfer enhancement can be achieved by placing the heat sources



between center to center. Chen et al. [2] performed experiments to determine the spacing effect for five protruding heat sources mounted on a horizontal channel, under forced convection. They found 12% drop in maximum temperature excess for optimum spacing in comparison to equally spaced heat sources. Durgam et al. [3] numerically studied the optimal arrangement of heat sources of rectangular geometry mounted on a PCB board using COMSOL MULTIPHYSICS under natural and forced laminar convection. They suggested that rate of heat transfer is enhanced using optimum spacing arrangement. Faraji and El Qarnia [4] presented the numerical results for cooling of three protruding discretely heated elements mounted on a conducting vertical plate, and placed in a rectangular chamber under the natural convection, and suggested that electronic components at the bottom of the cavity improve their cooling performance. Habib et al. [5] experimentally conducted the natural convection flow over an array of flush mounted discrete heat sources on a vertical substrate for both uniform and non-uniform spacing, and found that non-uniform spacing gives a better rate of heat transfer. Leung and Kang [6] performed experimental and numerical analysis to investigate the convective heat transfer in a rectangular duct using stream wise-periodic twelve identical rectangular heat sources mounted on the substrate board with uniform spacing of 6.35 mm between them, and found that top surface area and thickness of heat sources play a vital role in convective heat transfer, and most of the heat is dissipated from the top surface. Narasimham [7] made an overview of natural convection from discrete heat sources placed in an enclosure and suggested that effect of radiation on natural convection can be neglected. Pirasaci & Sivrioglu [8] conducted experiments on mixed convection heat transfer on top and bottom heated horizontal channel with arrays of protruded discrete heat sources of different height/width ratios, and found that, for lower and higher values of Richardson number heat transfer is natural convection and forced convection respectively, and for  $0.2 \leq Ri \leq 12.5$  heat transfer is mixed convection. Sudhakar et al. [9] conducted combined experiment and numerical study to predict the optimality of heat sources distribution in a vertical duct by carrying out numerical simulations using FLUENT. Sultan [9] studied the forced convection cooling effects from multiple protruding heat sources mounted in a horizontal channel. Yadav and Kant [11] studied numerically and experimentally the array size effect and substrate thermal conditions on heat transfer rate from heated modules using k- $\epsilon$  model. Hotta et al. [12] studied the radiation effect on the optimal arrangement of discrete heat sources under natural convection and suggested surface radiation is significant for cooling of discrete heat sources. Hotta and Venkateshan [13] studied the optimal arrangement of discrete heat sources using ANN-GA based methodology under the natural convection and concluded that combined ANN-GA based optimization technique gives more accurate results than conventional heuristic method. Hotta et al. [14] conducted ANN-GA based methodology drive by experimental data to obtain an optimal arrangement of discrete heat sources under the mixed convection and found that heat transfer is enhanced by painting the surfaces of heat sources black, which reduces temperature by 15%.

From the literature, it is seen that many studies have focused on identical protruding heat sources mounted with uniform spacing under natural and forced convection. The numerical analysis has been carried using ANSYS FLUENT and COMSOL MULTIPHYSICS. It is clear that none of the researchers have used ANSYS Icepak which uses object based modeling. Also, not many researchers have focused on mixed convection cooling considering the effect of surface radiation. The present work focuses on the numerical analysis of natural and mixed convection heat transfer on the optimal arrangement of five non-identical protruding heat sources (Aluminium) mounted on a PCB board (Bakelite) using ANSYS Icepak considering the effect of surface radiation.

### 3. Numerical Analysis

The numerical analysis is carried out using ANSYS Icepak for both natural and mixed convection. IcePak is object-based modeling software having huge application to the electronics industry. The model used in present study consists of the test section (Cabinet), the substrate (Bakelite) board, and five non-identical rectangular heat sources (Aluminium) mounted on it, as shown in Figure. 1.

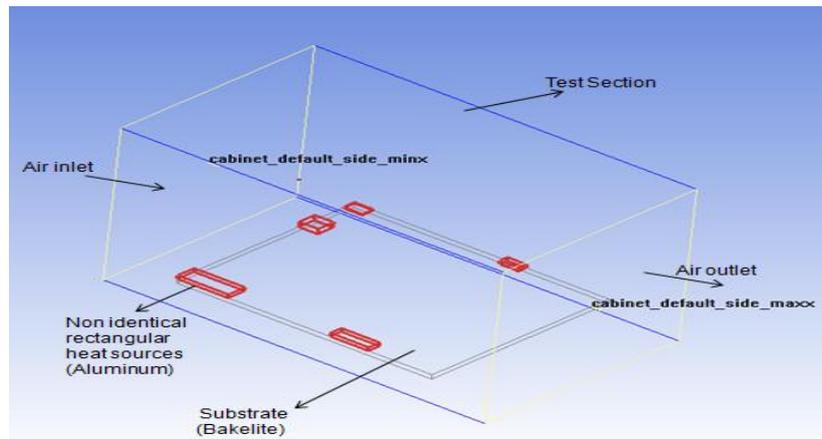


Figure 1. Computational model for mixed convection.

The optimal distribution of these heat sources have been determined by Hotta and Venkateshan [13] for natural convection, and by Hotta et al. [14] for mixed convection heat transfer. The experiments are also carried out by them for the optimal configuration of heat sources, to find the temperature distribution, and to discuss the thermal interaction between heat sources. The present paper deals with the numerical investigation part. Figure 2 shows the optimal configuration under natural and mixed convection heat transfer as obtained by [13] and [14].

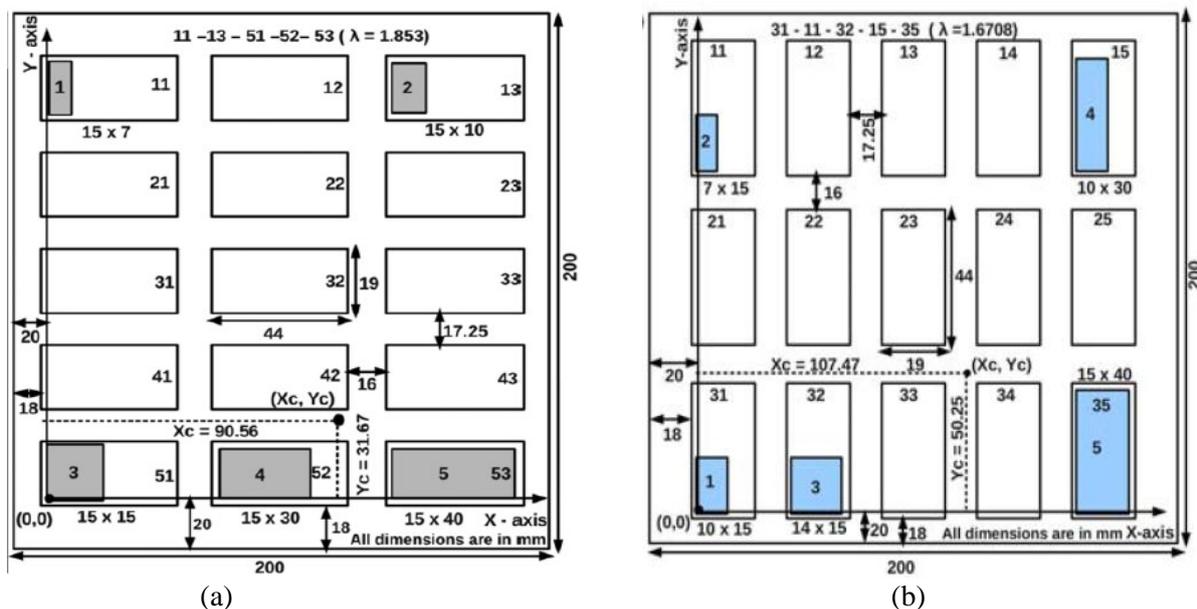


Figure 2. Optimal configuration of five heat sources (a) Natural convection. (b) Mixed convection.

### 3.1 Governing Equations and Boundary conditions

ANSYS Icepak solves the Navier-Stokes equations. The continuity equation can be written as

$$\frac{\partial \rho}{\partial t} + \nabla(\rho \vec{v}) = 0 \tag{1}$$

For an incompressible fluid, this reduces to,  $\nabla \cdot \vec{v} = 0$ .

Transport momentum equations in an inertial (non-accelerating) reference frame is

$$\frac{\partial}{\partial t}(\rho\vec{v}) + \nabla \cdot (\rho\vec{v}\vec{v}) = -\nabla p + \nabla \cdot (\bar{\tau}) + \rho\vec{g} + \vec{F} \quad (2)$$

Where  $p$  is the static pressure,  $\bar{\tau}$  is the stress tensor and  $\rho\vec{g}$  is the gravitational body force.  $\vec{F}$  contains other source terms due to resistance. The energy equation for a fluid region can be written in terms of sensible enthalpy  $h$ , where  $T_{ref}$  is 298.5K

$$h = \int_{T_{ref}}^T C_p dT \quad (3)$$

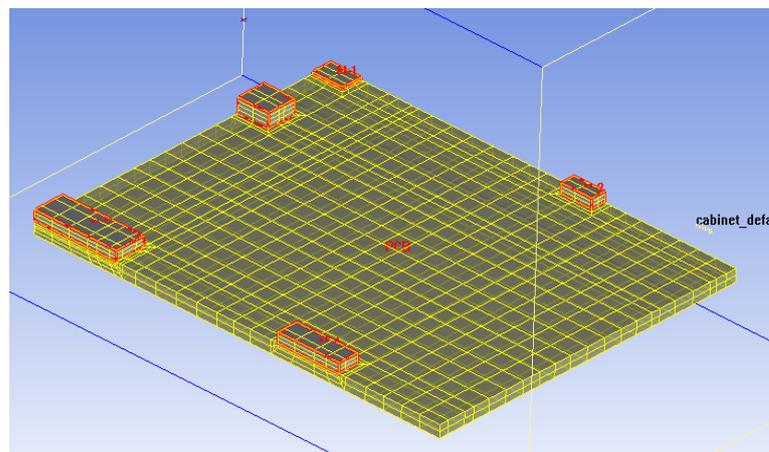
$$\frac{\partial}{\partial t}(\rho h) + \nabla \cdot (\rho h \vec{v}) = \nabla \cdot [(k + k_t)\nabla T] + S_h \quad (4)$$

Where  $k$  is the molecular conductivity,  $k_t$  is the turbulent transport conductivity ( $k_t = c_p \mu_t / Pr_t$ ) and  $T$  is temperature and the source term  $S_h$  is the volumetric heat source. The boundary conditions for natural convection (vertical orientation of substrate board, air flows in vertical direction against gravity) is taken as ambient temperature at  $Y$  (vertical axis) = 0,  $T = T_\infty = 25^\circ\text{C}$  and at  $Y = L$ ,  $p = p_\infty$ , whereas for mixed convection (horizontal orientation of substrate board) at  $X$  (horizontal axis) = 0,  $T = T_\infty = 25^\circ\text{C}$  and at  $X = L$ ,  $p = p_\infty$ , the velocity is 0.5m/s. The heat sources are subjected to a uniform heat flux of 2000 W/m<sup>2</sup>. The dimensions of the computational domain used in present study are

1. Substrate board: Length 175 mm, Width 175 mm and thickness 5 mm
2. Heat sources length, width, and thickness are taken for the optimal configuration as reported in [13] and [14].

### 3.2 Grid Independence study

ANSYS Icepak uses FLUENT solver which is based on control volume technique, the pre-processor generated HEXA dominant mesh which gives better results and has all the components of standard geometries. The computational domain consists of 27750 nodes, 25156 HEXA elements and 6504 quads for mixed convection, and for natural convection 138588 nodes, 130663 HEXA elements and 18736 quads. Increasing mesh elements beyond these values does not impact much on maximum temperature values. The maximum temperature difference between higher node elements is found 0.5°C. This strategy saves computational time and gives better results with respect to experimental values. The mesh profile for the optimal configuration under mixed convection is shown in Figure 3.



**Figure 3.** Mesh profile used in ANSYS Icepak for the optimal configuration under mixed convection.

Surface radiation modeling is enabled and view factor is computed using adaptive refined reference method for all the heated elements mounted on the substrate. The convergence criteria are set, and the simulations are carried for the optimal configuration under both natural and mixed convection heat transfer.

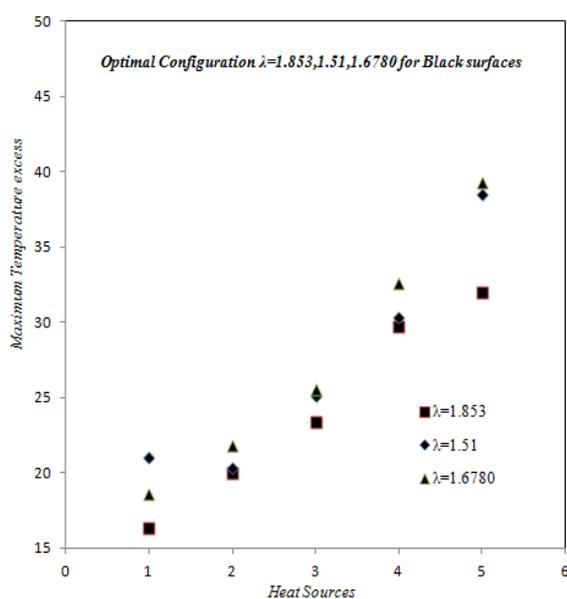
## 4. Results and Discussion

### 4.1 Natural Convection

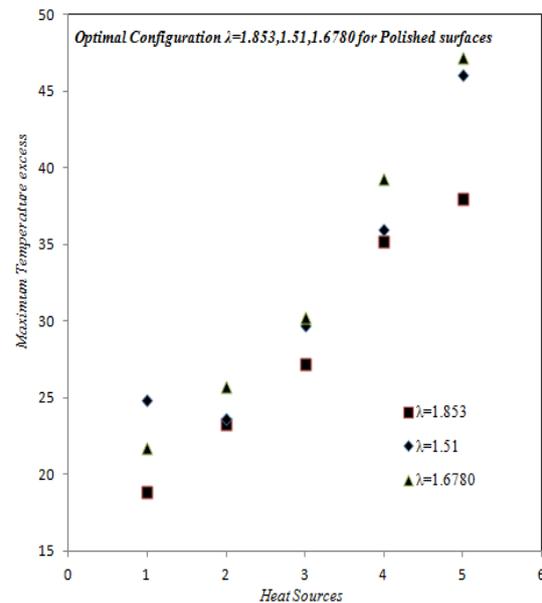
Five non identical rectangular heat sources were placed on the substrate. Multiple arrangements of heat sources are possible on substrate board and the optimal configuration is determined using heuristic non-dimensional geometric distance parameter  $\lambda$ , as reported in Hotta et al. [13]. Three configurations (on the basis of  $\lambda$  parameter, i.e, 1.51, 1.67, 1.851) are considered for simulation, by supplying uniform heat flux of  $2000\text{W/m}^2$  to all the heat sources, and for both polished and black surfaces.

#### 4.1.1 Maximum excess temperature variation for different arrangement

The maximum excess temperature variation among all the 5 heat sources for the above configurations is shown in Figure. 4 (black painted surface) and Figure. 5 (polished surface). It is seen from both the figures that, maximum temperature excess reduces as the configuration  $\lambda$  increases. Black painted surface losses maximum amount of heat by radiation in comparison to the polished surface. The percentage contribution of radiation is 6-13% and the temperature drop from polished to black painted surface is 6-12.5%, which matches with experimental values.



**Figure 4.** Variation of maximum temperature excess for black painted surface.

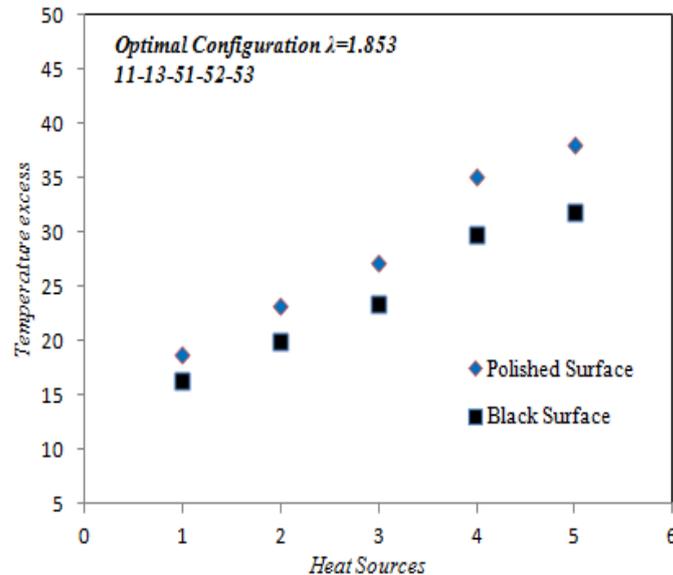


**Figure 5.** Variation of maximum temperature excess for polished surface

#### 4.1.2 Variation of maximum temperature excess for the optimal configuration, 11–13–51–52–53 ( $\lambda=1.853$ )

Figure 6 shows the maximum excess temperature for 5 heat sources of the optimal arrangement. It is seen that, the maximum temperature is obtained for heat source 5 (largest size), hence it has to be kept

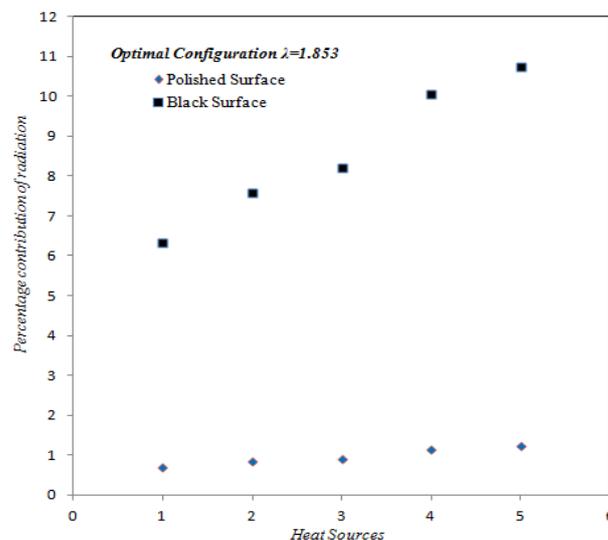
at 53 (position), substrate bottom corner. The temperature excess increases with the increase in the size of heat sources. The minimum temperature excess is for heat source 1 placed at 11 which is smallest among the five heat sources. Therefore it is clear that the heat source size has a greater influence on dissipation rate. There is a temperature drop of 6-10.6% by painting the surfaces with black, thus indicating the importance of radiation.



**Figure 6.** Excess temperature variation for 5 heat sources of the optimal arrangement.

#### 4.1.3 Percentage Contribution of Radiation for the optimal configuration 11–13–51–52–53 ( $\lambda=1.853$ )

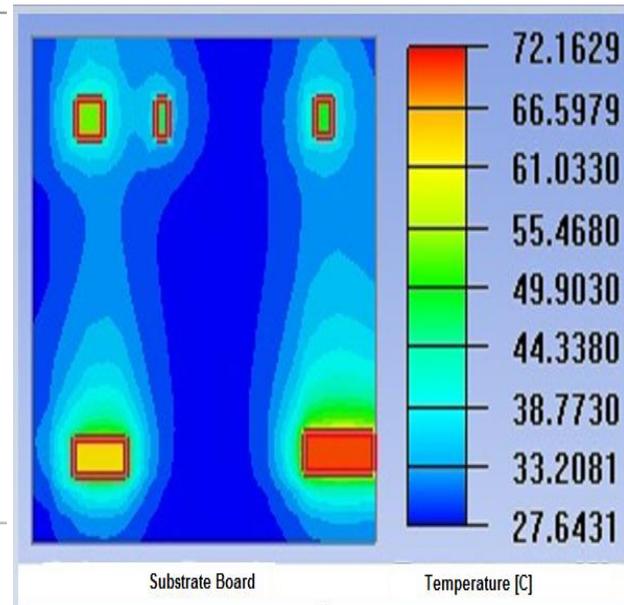
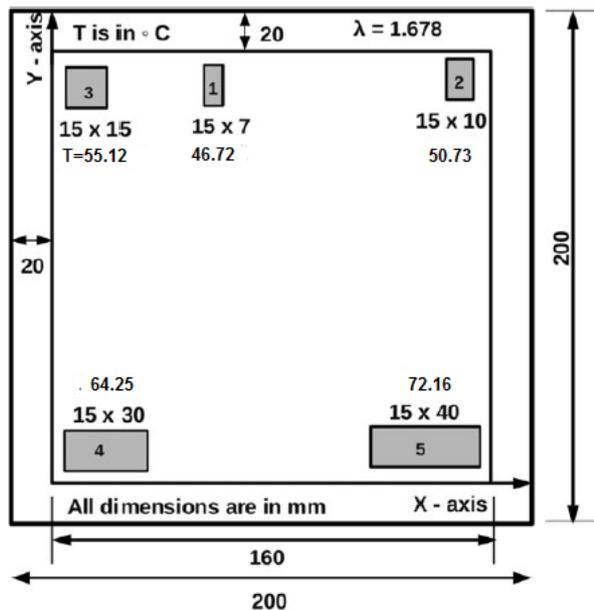
Figure 7 shows the contribution of surface radiation of 5 heat sources for the optimal arrangement. It is seen that the contribution of radiation for polished surface (emissivity  $\epsilon=0.08$ ) is low, i.e. 0.7 to 1.3%, whereas for black painted surface (emissivity  $\epsilon=0.85$ ) it is high, i.e. 6.7 to 11%. The contribution of radiation is minimum for heat source 1 positioned at the top corner and maximum for heat source 5 positioned at the bottom corner of the substrate.



**Figure 7.** Percentage contribution of radiation for heat Sources of optimal configuration.

#### 4.1.4 Temperature variation for five polished heat sources with $\lambda=1.678$

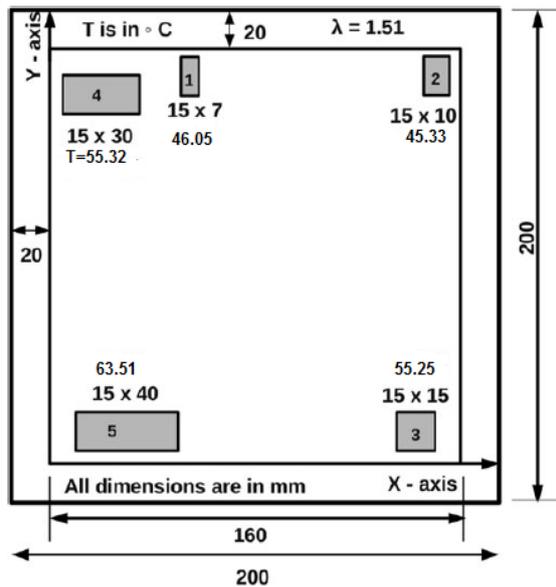
The optimal configuration for five polished non-identical heat sources which were generated using GA is reported by Hotta et al. [13]. The  $\lambda$  for this configuration is 1.678 which is less as compared to heuristic optimized value 1.853. Maximum temperature excess is obtained for heat source 5 placed at the bottom corner of the substrate, i.e. 47.16°C. This value is compared with the GA obtained from Hotta et al. [13] and the results strongly agree with an error of  $\pm 5\%$ . Figure 8 and 9 shows the position and temperature profile of five non-identical heat sources.



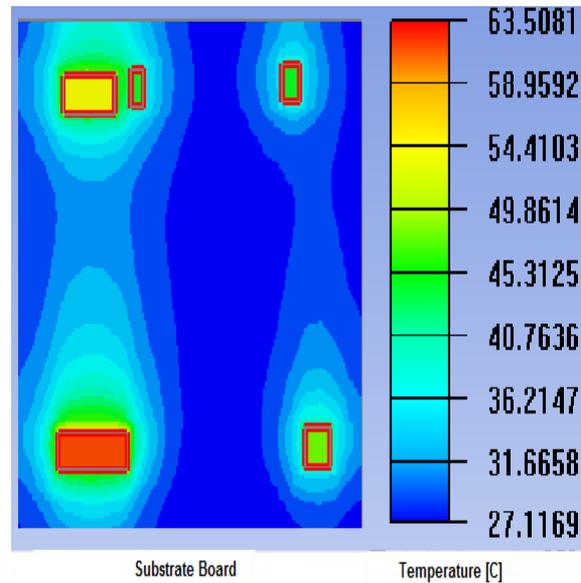
**Figure 8.** Position of five polished heat sources. **Figure 9.** Temperature contours five polished heat sources.

#### 4.1.5 Variation of temperature for five black painted heat sources with $\lambda=1.51$

Figure 10 shows the optimal configuration for five black painted non-identical heat sources ( $\lambda = 1.51$ ) generated using GA, as reported in Hotta et al. [13]. Maximum temperature excess of 38.15°C is obtained for heat source 5 placed at the bottom corner of the substrate. By changing positions of heated elements, the temperature of heat sources is greatly affected. Hence, the size of the heat sources plays a significant role in heat dissipation. This value compared with the GA obtained from Hotta et al. [13] and the results agree with an error of  $\pm 8.5\%$ . Figure 11 shows the temperature contour of five black painted heat sources.



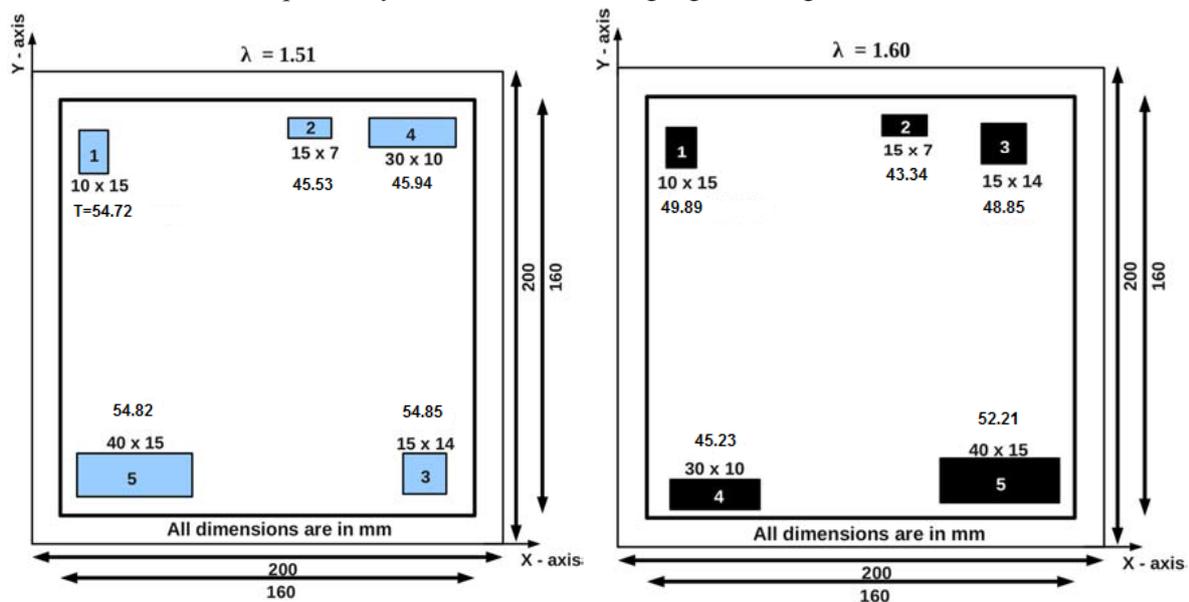
**Figure 10.** Position of five black painted heat sources.



**Figure 11.** Temperature contours of five black painted heat source.

4.2 Mixed Convection

Five non-identical rectangular heat sources were placed on the substrate. Multiple configurations of heat sources are possible on a substrate board, and the optimal configuration is determined using heuristic non-dimensional geometric distance parameter  $\lambda$ , reported in Hotta et al. [14], out of which three configurations are considered for simulation  $\lambda=1.50, 1.60, 1.6708$ . Simulations are performed by supplying uniform heat flux of  $2000 \text{ W/m}^2$  and velocity of  $0.5\text{m/s}$  to all the heat sources. The optimal configuration for polished and black painted heat sources (Figure. 12 (a) and (b)) is obtained for  $\lambda=1.51$  and  $1.670$ , as reported by Hotta et al. [14] using a genetic algorithm.



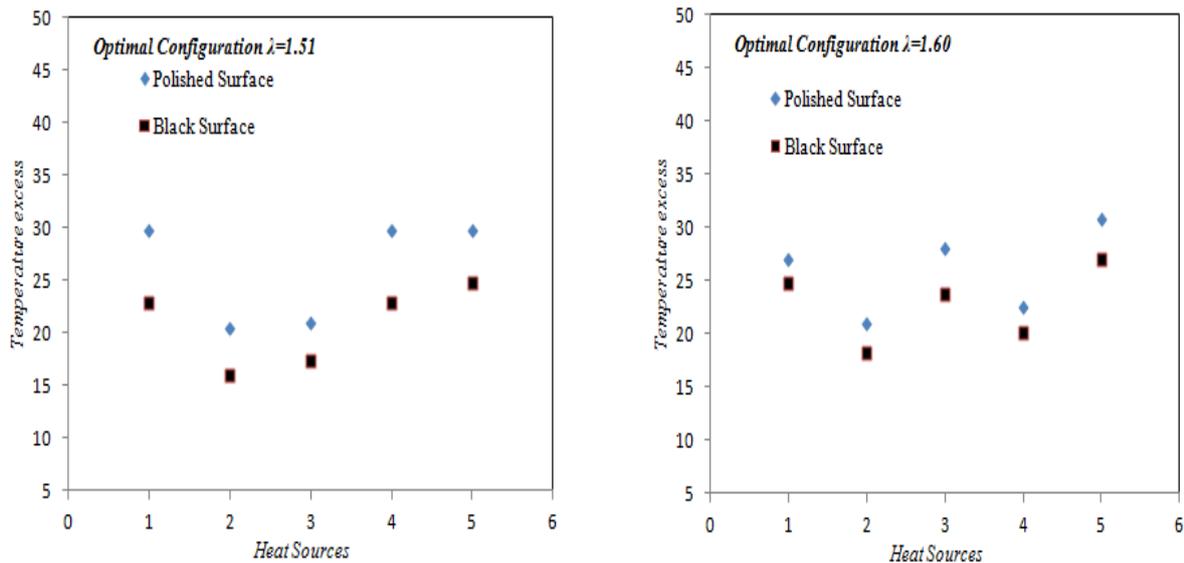
a) Polished surface.

b) Black painted surface.

**Figure 12.** Optimal configuration of five rectangular heat sources under mixed convection.

4.2.1 Variation of maximum temperature excess for optimal configuration  $\lambda=1.51$  and 1.60

Figure 13 (a) shows the maximum temperature excess for configuration  $\lambda = 1.51$ , and is seen that the temperature excess is maximum for heat source 5 and minimum for heat source 2. Hence, the size of the heat sources greatly affects the heat source temperature. The temperature drop from polished surface to black painted heat source is 8 to 15%, which confirms the role of radiation to be significant. Figure 13 (b) shows the maximum temperature excess for configuration  $\lambda = 1.60$  and found to be is 27.1° C for black painted heat sources which are higher than the configuration 1.51. The temperature drop from polished surface to black painted heat source is 4 to 7%.

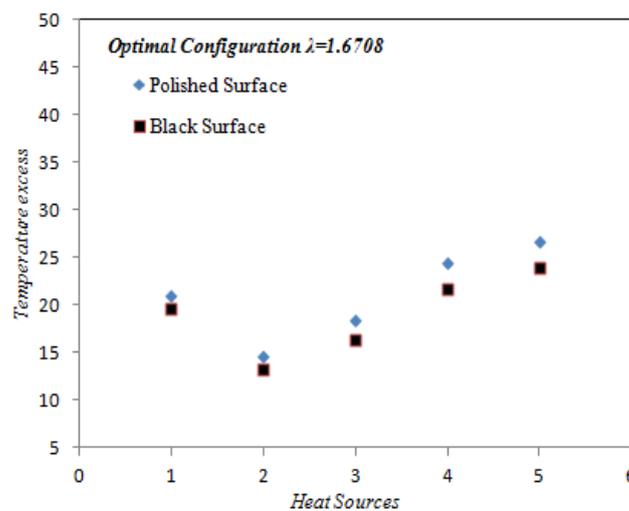


a)  $\lambda = 1.51$

b)  $\lambda = 1.60$

**Figure 13.** Variation of maximum temperature excess for different configuration.

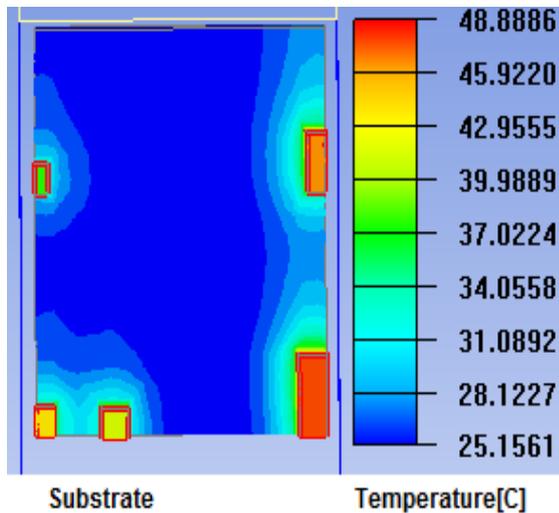
4.2.3 Variation of Maximum Temperature Excess for Optimal configuration  $\lambda=1.67$



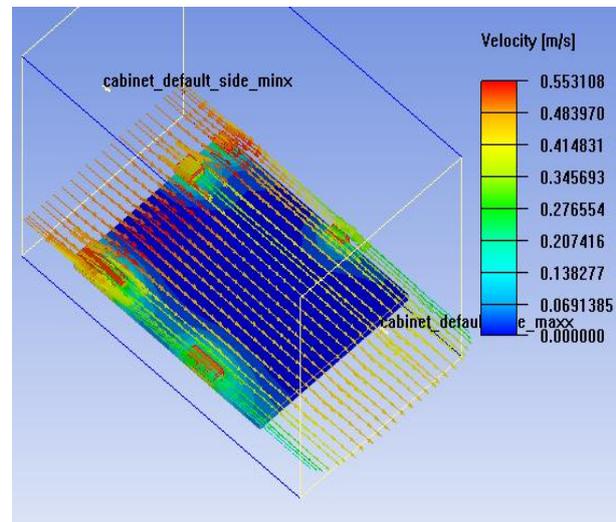
**Figure 14.** Excess temperature variation for 5 heat sources of optimal arrangement.

Figure 14 shows the maximum temperature excess for both black painted and polished surface, and the reported values are 23.89°C and 26.76 °C respectively. This maximum temperature excess is the

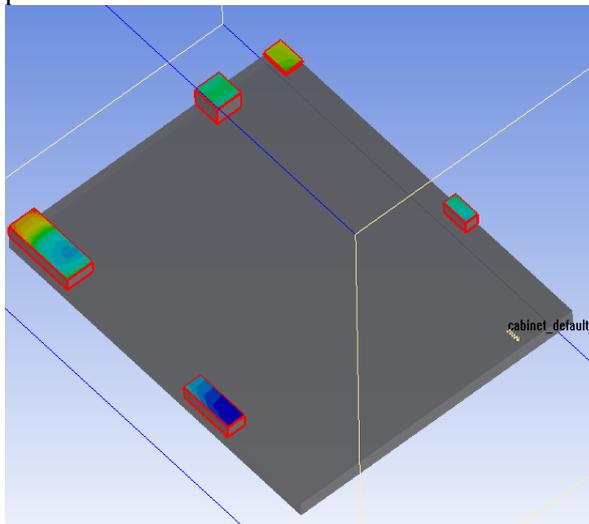
minimum for the configuration 1.51 and 1.60, which satisfies the objective in reducing the maximum temperature excess. The temperature drop from polished surface to black painted heat source is obtained is 3 to 6%.



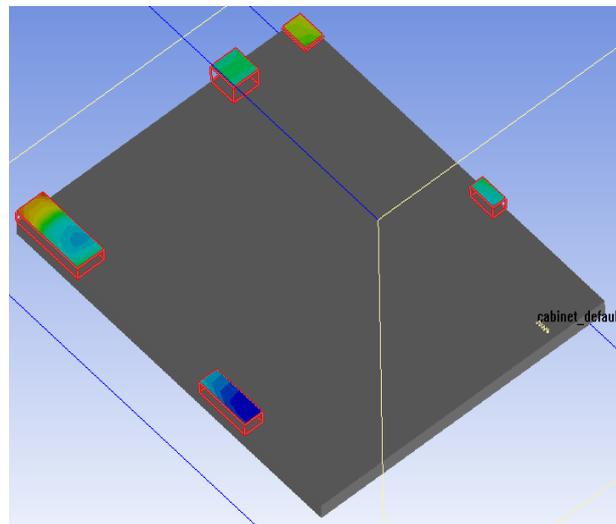
**Figure 15.** Temperature contours for black painted heat sources.



**Figure 16.** Velocity profile for black painted heat sources.



**Figure 17.** Heat transfer coefficient for black painted heat sources.



**Figure 18.** Heat transfer coefficient for polished heat sources.

## 5. Conclusion

Numerical analysis using ANSYS Icepack is conducted under both steady-state natural and mixed convection heat transfer experiments for 5 non-identical protruding rectangular heated elements mounted at different positions on a substrate board. Multiple configurations are possible for mounting 5 heat sources on the board. The optimal arrangement is based on  $\lambda$  for both natural convection and mixed convection heat transfer. The maximum temperature reduces with increase of  $\lambda$ . The temperature excess is maximum for heat source having the largest size. In natural convection heat transfer the maximum temperature excess is 38.1°C, whereas, for mixed convection, it is 23.89°C.

Radiation is significant, as there is a temperature drop of 6-13% under the natural convection, and 3-15% under mixed convection. The results suggest that mixed convection is a better method for cooling of discrete heat source modules. ANSYS Icepak can be used for predicting the temperature, velocity, pressure of heat sources mounted on a substrate which is in strong agreement with experimental values.

## 6. References

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