

Convective Heat Transfer on the Optimum Spacing of High Heat Dissipating Heat Sources—A Numerical Approach



K. Mathew and Naveen Patil

Abstract The paper emphasizes the numerical investigation to analyse the three modes of conjugate convective heat transfer characteristics from seven non-identical heat sources (Aluminium) mounted on a substrate board (Bakelite). The objective is to determine the best mode of heat transfer which suits for cooling of ICs (heat sources), also temperature is non-dimensionalized as (θ), and it is predicted by applying fuzzy logic control. To accomplish this, numerical simulations are carried out using ANSYS Icepak to estimate the temperature distribution of the IC chips. Results suggest that the mixed convection is the better mode of heat transfer and the temperature of heat sources is reduced. Temperature of the heat sources is a strong function of their size and position on the substrate board. There is a strong agreement between the numerical values of ANSYS Icepak and predicted values obtained from fuzzy logic.

Keywords CFD simulations · Fuzzy logic · Heat sources · Convective modes of heat transfer · Optimal arrangement

1 Introduction

Present-day electronic equipment is working under highly demanding conditions, with satisfactory performance depending on the operating temperature. Effective cooling is required to keep this equipment within safe temperature limits of 85 °C. For low and medium heat-generating devices, air cooling is widely used. Though natural convection cooling does not require additional devices to create air movement, the rate of heat removal is much lower. Forced convection cooling is preferred whenever high heat transfer rates are required. It confirms from the literature that, the various factors responsible for the failure of electronic components [1] are humidity (16%), vibration (24%), dust (6%) and temperature (54%). This clearly indicates that

K. Mathew (✉)

MIT School of Engineering, MIT-ADT University, Pune, India

e-mail: mathew.karvinkoppa@mituniversity.edu.in

N. Patil

Vellore Institute of Technology, Vellore, Tamil Nadu, India

© Springer Nature Singapore Pte Ltd. 2020

L. Vijayaraghavan et al. (eds.), *Emerging Trends in Mechanical Engineering*,

Lecture Notes in Mechanical Engineering,

https://doi.org/10.1007/978-981-32-9931-3_8

temperature leads the chart for the failure of electronic equipment and must have to be controlled with great interest. The electronic industries are facing a stiff challenge for maintaining the component temperature below 85 °C.

2 Background Study

Due to the rapid development of an integrated circuit technology, miniaturization will lead to gradual increase in circuit densities, and rate of heat dissipation increases tremendously; hence, the cooling of electronic components has become challenging. As the extent of electronic parts ends up littler, the area accessible for heat dissipation is reducing which leads to more challenging effective cooling systems. Numerical investigation was conducted to estimate the optimize positions of ICs on the SMPS board under mixed convection [2]. They found that the temperature variation of the ICs is a strong function of their size, position and heat flux. They proposed a correlation for these parameters. The optimum distribution of discrete heat sources was determined experimentally [3]. They used a robust hybridization technique (ANN and GA) to determine the optimal configuration of ICs and also they suggested that this will give better results than regular optimization techniques. Analysed experimentally and numerically forced convection cooling of ICs arranged optimally on the substrate board [4]. They suggest a correlated non-dimensional temperature (θ) in terms of Reynolds number (Re) and thermal conductivity (K). Steady-state experiments were conducted on five non-identical protruding discrete ICs under natural, mixed convection heat transfer medium [5]. Their goal is to find the rate of heat dissipation from ICs, keeping at the lowest bottom of the PCB and to compare both modes of heat transfer for cooling of heat sources. A complete review of the cooling on discrete heated modules under a different mode of heat transfer [6]. They suggested the choice of working fluid based on the operating level of heat flux ranges. Numerical investigation has been carried out to determine the three-dimensional natural convective heat transfer on flush mounted in the horizontal cavity filled with air [7]. They concluded the walls at the outer side are strongly affected by the convection. Studied the cooling performance of multiple protruding ICs mounted in the horizontal channel under forced convection, they found that heat transfer is maximum by 33.15% at Re 3428 [8]. They proposed the correlation for the average Nusselt number and the dimensionless temperature is maximum which is a function of the Richardson number. Optimal arrangement of rectangular ICs mounted on PCB is studied experimentally and numerically [9]. They used COMSOL MULTIPHYSICS and suggested a correlation for dimensionless temperature (θ) in terms of heat flux (q_{max}), geometric distance parameter (λ) and thermal conductivity of the substrate board (K_{the}), and it is observed that heat transfer rate is increased by arranging the ICs with optimal spacing. To estimate the array size and condition of heat substrate board with heat sources on the enhancement of rate, heat transfer was studied numerically and experimentally [10]. They proposed an empirical relation for evaluating the Nusselt number under different thermal conditions for array substrate board geometries.

From glance of literature review, many of the work has been focused on the single mode of heat transfer [2, 7, 9] and some of the studies were pertinent two modes of heat transfer [5, 6, 10] for the cooling of discrete heated modules, and ANSYS FLUENT and COMSOL MULTIPHYSICS are used for analysis of ICs. However, is clear that numerical analysis using ANSYS Icepak scarres. Hence, the present analysis emphasizes on numerical simulation on optimal configuration of non-dimensional geometric parameter distance $\lambda = 1.3$ under three different modes of conjugate convective heat transfer (free, forced and mixed convection). Fuzzy logic control is used to predict the temperature of heat sources (ICs). This will give a clear insight into the electronic cooling engineers regarding the selection of the mode of convective heat transfer for the cooling of electronic components. Hence, the study is critical for the thermal management designers.

3 Methodology

3.1 Numerical Simulation

Numerical simulation is carried out using ANSYS Icepak (R-16) for cooling of seven non-identical protruded ICs mounted on a substrate board under all the three-mode convective heat transfer (free, forced and mixed convection) for optimal configuration $\lambda = 1.3$ as reported in [9]. The ANSYS Icepak is an object-based modelling software with pre-defined electronic components and has specific applications in the electronics industry. Once the modelling is done, the Icepak uses FLUENT solver for the thermal and fluid flow calculation. However, all the set-ups are carried out using Icepak, and then the Icepak uses FLUENT to solve the governing equation such as the conservation equations of mass, momentum, energy and other scalars such as turbulence if needed. The FLUENT solver uses the schemes called control volume technique.

The dimensions of all the components for $\lambda = 1.3$ are depicted in Figs. 1 and 2. The heat sources are denoted by $AI_{1, \text{and}}$ the subscript stands for the position of heat sources.

Governing Equation

ANSYS Icepak solves the Navier–Stokes equations for transport of mass, momentum and energy for the laminar fluid flow and heat transfer. The equations are written as follows

The conservation equation of mass is given in Eq. 1.

$$\frac{\partial \rho}{\partial t} + \frac{\partial(\rho u)}{\partial x} + \frac{\partial(\rho v)}{\partial y} + \frac{\partial(\rho w)}{\partial z} = 0 \quad (1)$$

For an incompressible fluid, the value of density ρ is constant, and Eq. 1 reduces to the form Eq. 2.

Fig. 1 Components used for present computational study

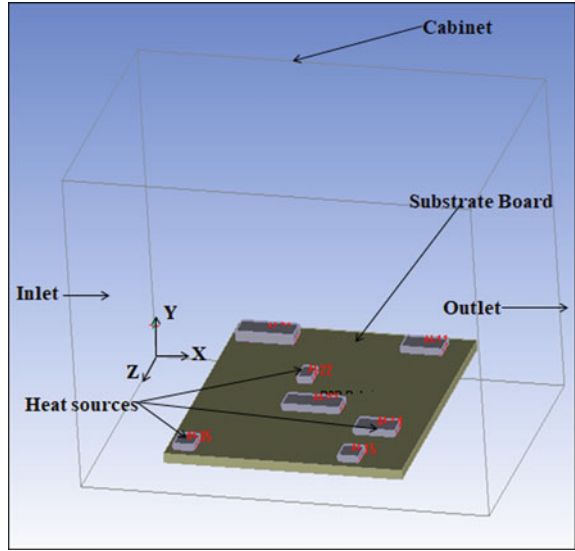
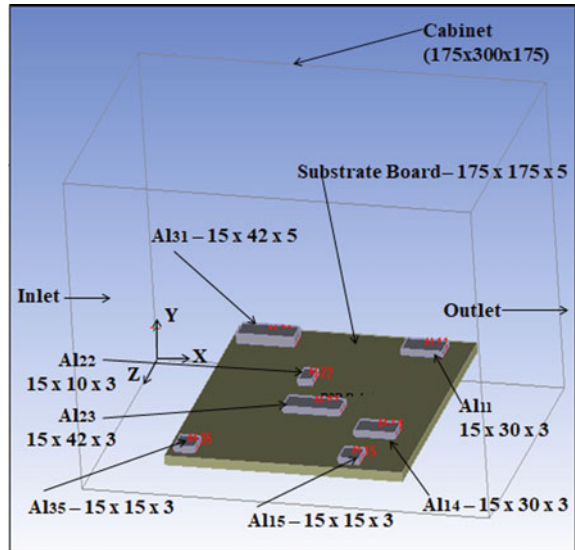


Fig. 2 Dimensions and locations of ICs (all dimensions are in mm)



$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0 \tag{2}$$

The momentum principle describes the relationship between the velocity, pressure and density of a moving fluid. The general form of the momentum equation (Navier-Stokes equation) for natural convection is given in Eqs. [3-5].

$$\frac{\partial u}{\partial t} + u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} + w \frac{\partial u}{\partial z} = -\frac{1}{\rho} \frac{\partial p}{\partial x} + \nu \left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2} \right) + \rho \beta (T_{\max} - T_{\text{amb}}) \quad (3)$$

$$\frac{\partial v}{\partial t} + u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} + w \frac{\partial v}{\partial z} = -\frac{1}{\rho} \frac{\partial p}{\partial y} + \nu \left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial z^2} \right) + \rho \beta (T_{\max} - T_{\text{amb}}) \quad (4)$$

$$\frac{\partial w}{\partial t} + u \frac{\partial w}{\partial x} + v \frac{\partial w}{\partial y} + w \frac{\partial w}{\partial z} = -\frac{1}{\rho} \frac{\partial p}{\partial z} + \nu \left(\frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2} + \frac{\partial^2 w}{\partial z^2} \right) + \rho \beta (T_{\max} - T_{\text{amb}}) \quad (5)$$

The above equation (Eq. 3–5), the buoyancy term $[\rho \beta (T_{\max} - T_{\text{amb}})]$, is zero for the forced convection case. Applying the first law of thermodynamics to a small control volume of an incompressible fluid, the final form of the energy equation in 3D can be written, as given in Eq. 6.

$$\frac{\partial T}{\partial t} + u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} + w \frac{\partial T}{\partial z} = \frac{k}{\rho c} \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} \right) + q_g + \emptyset \quad (6)$$

where q_g is the heat generation, C is the specific heat and \emptyset is the viscous dissipation, which is further written in the form as given in Eq. 7.

$$\emptyset = 2\mu \left[\left(\frac{\partial u}{\partial x} \right)^2 + \left(\frac{\partial v}{\partial y} \right)^2 + \left(\frac{\partial w}{\partial z} \right)^2 \right] + \mu \left[\left(\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} \right)^2 + \left(\frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} \right)^2 + \left(\frac{\partial w}{\partial z} + \frac{\partial u}{\partial x} \right)^2 \right] \quad (7)$$

For natural convection, driving force for working fluid is due to the difference in density. The boundary conditions for the natural convection, forced convection and mixed convection are mentioned in Table 1. The mixed convection velocity is calculated (Appendix) in such a way that, the maximum temperature of the ICs should not exceed 85 °C.

Table 1 Boundary conditions for different modes

Locations	Natural convection	Mixed convection	Forced convection
Inlet, $X = 0$	$V = 0.1 \text{ m/s}$ $T = T_{\infty} = 29 \text{ }^{\circ}\text{C}$ $\frac{\partial T}{\partial y} = \frac{\partial T}{\partial z} = 0$	$V = 0.78 \text{ m/s}$ $T = T_{\infty} = 29 \text{ }^{\circ}\text{C}$ $\frac{\partial T}{\partial y} = \frac{\partial T}{\partial z} = 0$	$V = 0.45 \text{ m/s}$ $T = T_{\infty} = 29 \text{ }^{\circ}\text{C}$ $\frac{\partial T}{\partial y} = \frac{\partial T}{\partial z} = 0$
Outlet, $X = L$	$p = p_{\infty}$, lateral boundary conditions are adiabatic		

Table 2 Mesh independency study

S. No.	Nodes	Hexa	Quads	T_{max} (°C)
1	19,348	17,296	533	56.84
2	11,836	10,388	3859	57.11
3	29,660	26,890	7029	56.31
4	89,532	83,540	15,170	56.45

Table 3 Validation of experimental and numerical results

$\lambda = 1.3$	Temperature °C	$q = 1500 \text{ W/m}^2$		$q = 2000 \text{ W/m}^2$	
		Experimental	Numerical	Experimental	Numerical
Natural convection	$T_{excess,max}$	36.6	39.85	50	50.61
Forced convection $V = 0.45 \text{ m/s}$	$T_{excess,max}$	26.5	23.61	30	31.48

Mesh Sensibility Study

The grid independence study has been carried out for ($\lambda = 1.3$) on basis keeping the temperature limit 85 °C. The grid independence study for configuration $\lambda = 1.3$ is given in Table 2. The grid sensitivity study is done with reference to Roache [11] for the present study. The grid size (Nodes, Hexas and Quads) is selected on the basis of the convergence criteria for mass, momentum (in x -, y - and z -direction) as well as energy balance equation. The safe temperature limit for the heat sources, i.e 85 °C is also kept in mind during the grid independence study. Hence, Table 3 represents the mesh independence of the based that 29,660 Nodes, Hexa 26,890 and Quads 7029 are selected for simulation. The convergence criteria $1e^{-6}$ for energy equation and $1e^{-6}$ for continuity and momentum equations are set. The mesh profile for mixed convection mode of heat transfer for configuration with $\lambda = 1.3$ is shown in Fig. 3.

3.2 Fuzzy Logic

In this study, the fuzzy system has been utilized keeping in mind the end goal to show and anticipate the numerical outcomes. A basic fuzzy comprises four noteworthy parts: Fuzzy interface, rules, and defuzzification interface. It has the impact of changing fresh information into fuzzy sets and connects between input and output with rules. For example, if x is A then y is B . Each fuzzy set is described by appropriate membership function. In the present investigation, Mamdani fuzzy modelled is used with the sources of inputs are λ , k^* , q^* and Re and yield θ as output. The defuzzification interface blends and changes over fuzzy sets into noteworthy numerical yields (Fig. 4).

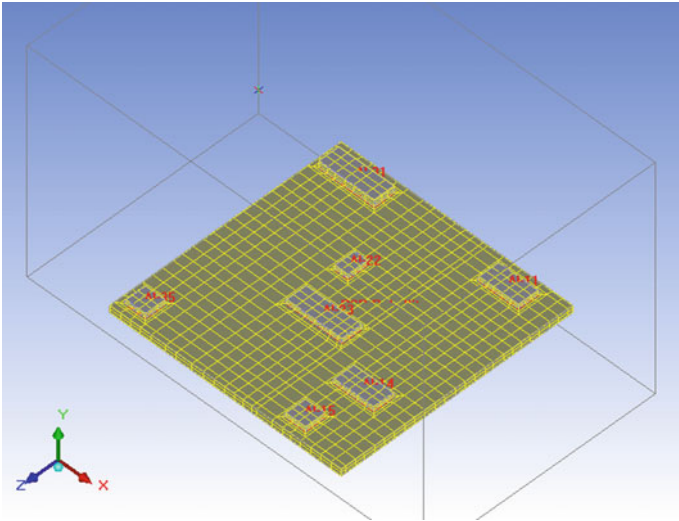


Fig. 3 Mesh sensibility for configuration $\lambda = 1.3$

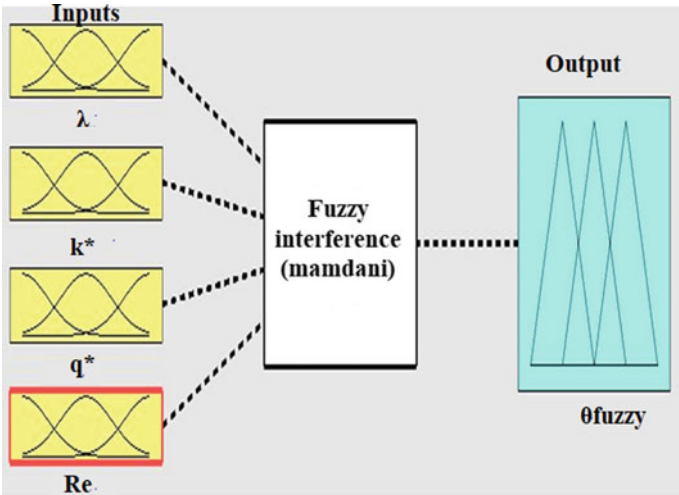


Fig. 4 Fuzzy inference system with inputs and output

4 Result and Discussion

4.1 Validation of Numerical Results

Seven non-identical rectangular ICs are placed on the substrate board (Bakelite). Diverse blend is workable for keeping ICs on the substrate board and the optimal set-up is obtained utilizing heuristic non-dimensional geometric separation parameter λ , as revealed in [9]. Numerical analysis is carried on optimal configuration $\lambda = 1.3$ for three modes of convective heat transfer (free, forced and mixed convection) by supplying constant heat flux of $q = 1500 \text{ W/m}^2$ and $q = 2000 \text{ W/m}^2$. It was observed that there is a strong agreement between the experimental values and numerical values for natural convection and forced convection as shown in the table. The experimental values were obtained from [9].

Figure 5 shows the temperature distribution of heat sources for optimal configuration $\lambda = 1.3$ for three modes of convective heat transfer. Figure 5a depicts the temperature of heat sources for constant heat flux of 1500 W/m^2 . It is clearly seen that for natural (free) convection heat transfer, the temperature of all heat sources is maximum and minimum for mixed convection heat transfer. Heat source number 2, located at 11, has the maximum temperature among all heat sources for all three-mode convective heat transfer. The maximum temperature of the heat source is $49.48 \text{ }^\circ\text{C}$ for mixed convection, which is the minimum in comparison with natural and forced convection as shown in Fig. 5a. In a similar manner, Fig. 5b shows the temperature of heat sources for constant heat flux of 2000 W/m^2 . The maximum temperature of the heat source is $53.59 \text{ }^\circ\text{C}$ for mixed convection located at 11 positions, which is the minimum in comparison with natural and forced convection as shown in Fig. 5b. From Fig. 5a and b, it is evident that the heat sources with maximum temperature is placed at the lower edge of the substrate board stating further that the temperature of heat sources is minimum for mixed convection heat transfer in both

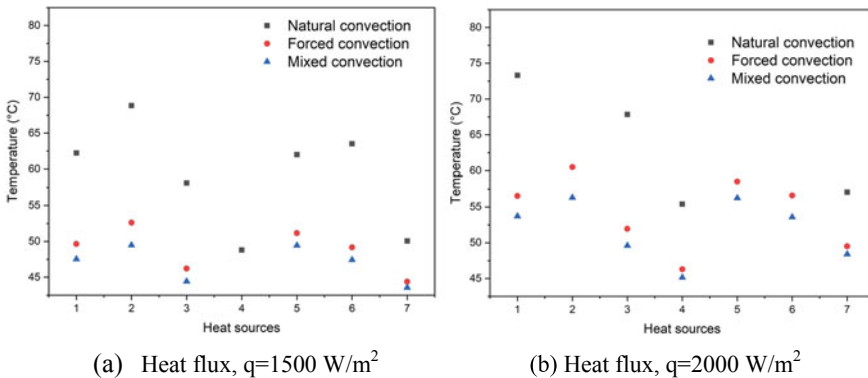


Fig. 5 Temperature of heat sources for optimal configuration $\lambda = 1.3$

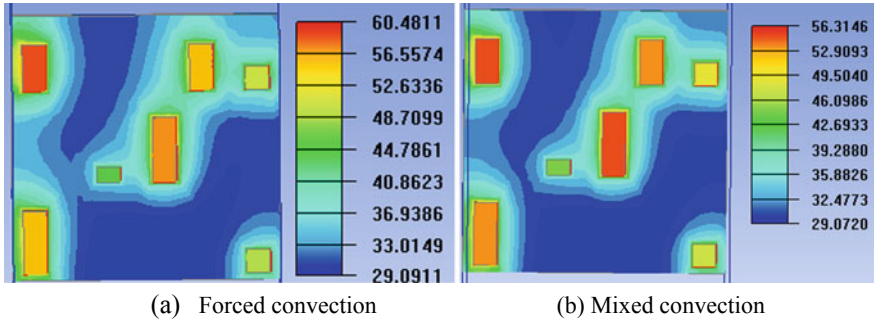


Fig. 6 Temperature contours of ICs for $q = 2000 \text{ W/m}^2$

the case of constant heat flux supplied of 1500 and 2000 W/m^2 . Figure 6a and b shows the temperature distribution for forced convection and mixed convection heat transfer with constant heat flux 2000 W/m^2 , respectively.

Fuzzy logic control is applied to the forced convection correlation proposed by Durgam et al. [9] for non-dimensional temperature, which is function of non-dimensional heat flux (q^*), non-dimensional thermal conductivity (k^*) and non-dimensional geometric parameter distance (λ), given in Eq. 8

$$\theta = 8.34 \times 10^{-6} q^* + 0.11 k^* - 0.35 \lambda \tag{8}$$

Equation 8 is valid for the range of parameters, $0.41 \leq \lambda \leq 1.63$, $11.21 \leq k^* \leq 330$, $1176 \leq q^* \leq 76,542$ and $1125 \leq \text{Re} \leq 2625$. The values of all the parameters (λ , k^* , q^* and Re) are interpolated for 47 values between the range at equal intervals and the non-dimensional temperature correlation (θ_{corr}) is calculated. The rules of the fuzzy logic control are defined on the basis of the variation of all independent variables on θ_{corr} as shown in Fig. 7. Figure 8 shows the parity plot of $\pm 12\%$ between θ_{corr} and predicted values of θ_{fuzzy} .

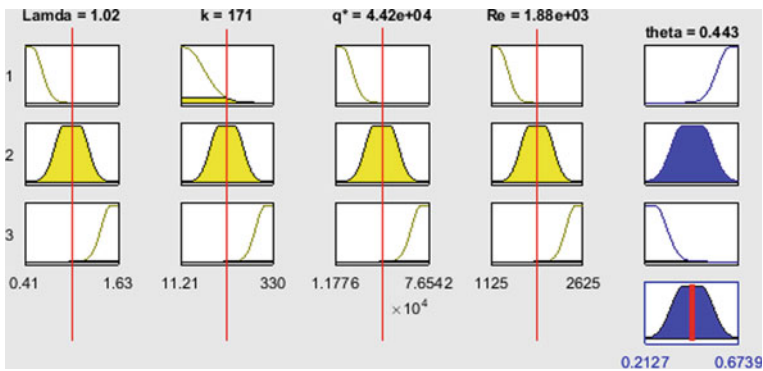


Fig. 7 Rule chart for predicting non-dimensional (θ_{fuzzy})

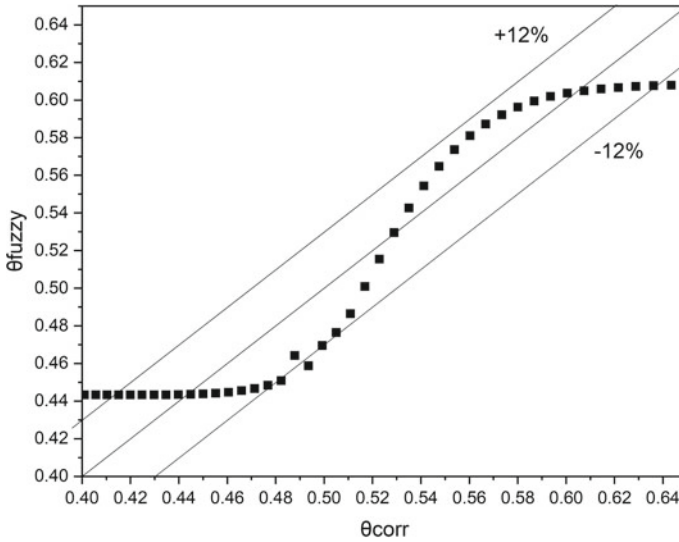


Fig. 8 Parity plot, showing the agreement between temperature θ_{corr} and θ_{fuzzy}

5 Conclusion

A numerical investigation for natural, forced and mixed convection heat transfer on seven non-identical heat sources (Aluminium) mounted on substrate board (Bakelite) is conducted on the optimal configuration of $\lambda = 1.3$ as reported in the literature. It is observed that mixed convection heat transfer as the minimum of maximum temperature of ICs of 53.59 °C as compared to natural and forced convection heat transfer. It is also seen that the ICs with maximum temperature should be placed at the lower side of substrate, which results in more interaction of heat source with cooling air. The experiment results from literature are in strong agreement with the numerical results. An attempt is made to apply fuzzy logic control to predict the temperature of heat sources for forced convection. There is strong agreement between the numerical values of ANSYS Icepak and predicted values obtained from fuzzy logic with $\pm 12\%$ error. Fuzzy logic is an innovative way to predict the temperature of ICs for the electronic cooling engineers and the selection of mode of convective heat transfer for the cooling of electronic components. Hence, the study is critical for the thermal management designers.

Appendix

$$1 = \frac{Gr}{Re^2} = Ri$$

$$Gr = \frac{g\beta\Delta T_{ref}H^3}{\nu^2}, Re = \frac{uL_h}{\nu}$$

$$\beta = \frac{1}{T_{mean}} = \frac{1}{(55 + 273)} = 3.04878 \times 10^{-3} K^{-1},$$

$$L_h = \frac{4A}{P} = \frac{4 \times (0.042 \times 0.015)}{2(0.042 + 0.015)} = 22.1052 \times 10^{-3} m$$

$$\Delta T_{ref} = \frac{qL}{K_{Al}} = \frac{2000 \times 0.042}{230} = 0.3652 K$$

$$1 = \frac{9.81 \times 3.04878 \times 10^{-3} \times 0.365 \times 0.3^3}{(u \times 22.105 \times 10^{-3})^2}$$

$$u = 0.7768 \approx 0.78 \text{ m/s}$$

References

1. Reynell M (1990) Major causes for failure of electronic components. U.S. Air Force Avionics Integrity Program
2. Mathew VK, Hotta TK (2018) A Numerical investigation on the optimal arrangement of IC chips mounted on an SMPS board cooled under mixed convection. *Therm Sci Eng Prog* 7:221–229
3. Hotta TK, Balaji C, Venkateshan SP (2015) Experiment driven ANN-GA based technique for optimal distribution of discrete heat sources under mixed convection. *Exp Heat Transf* 28:298–315
4. Durgam S, Venkateshan SP, Sundararajan T (2018) A novel concept of discrete heat source array with dummy components cooled by forced convection in a vertical channel. *Appl Therm Eng* 129:979–994
5. Hotta TK, Venkateshan SP (2012) Natural and mixed convection heat transfer cooling of discrete heat sources placed near the bottom on a PCB. *Int J Mech Aersp Eng* 6:266–273
6. Patil NG, Hotta TK (2018) A review on cooling of discrete heated modules using liquid jet impingement. *Front Heat Mass Transf* 11
7. Gavara M, Kanna PR (2014) A three-dimensional study of natural convection in a horizontal channel with discrete heaters on one of its vertical walls. *Heat Transf Eng* 35:1235–1245
8. Sultan GI (2000) Enhancing forced convection heat transfer from multiple protruding heat sources simulating electronic components in a horizontal channel by passive cooling. *Micro-electr J* 31:773–779
9. Durgam S, Venkateshan SP, Sundararajan T (2017) Experimental and numerical investigations on optimal distribution of heat source array under natural and forced convection in a horizontal channel. *Int J Therm Sci* 115:125–138

10. Yadav V, Kant K (2007) Air cooling of variable array of heated modules in a vertical channel. *J Electron Packag* 129:205
11. Roache P, Perspective J. A method for uniform reporting of grid refinement studies. *J Fluid Eng* 116: 405–413