Sādhanā (2018) 43:187
https://doi.org/10.1007/s12046-018-0950-7

Role of working fluids on the cooling of discrete heated modules: a numerical approach

NAVEEN G PATIL and TAPANO KUMAR HOTTA*

School of Mechanical Engineering, Vellore Institute of Technology (VIT), Vellore 632014, India
e-mail: naveenpatil.g@vit.ac.in; tapano.hotta@vit.ac.in

MS received 23 November 2017; revised 29 March 2018; accepted 2 May 2018; published online 9 October 2018

Abstract. The study has focused on the role of working fluids (air, water and FC-72) on the cooling of discrete heated modules under free, forced and mixed convection medium. Three non-identical protruding discrete heat sources are arranged at different positions on a substrate board following golden mean ratio (GMR). Numerical simulations for these heat sources are carried out using a commercial software (ANSYS-Icepak R-15) to simulate their flow and temperature fields under three different modes of heat transfer. Results suggest that the temperature of the heat sources is a strong function of their size, position on the substrate board, the velocity of the fluid and type of working fluid used. A correlation has been proposed for the temperature of these heat sources keeping in mind their strong dependence on the afore-mentioned parameters. It has been found that water can be used for better heat removal from the heat sources due to its high boiling point. The whole idea gives a clear insight to the electronic cooling engineers regarding the selection of working fluids and modes of heat transfer for the cooling of electronic components.

Keywords. ANSYS-Icepak; discrete heat source; forced convection; mixed convection; natural convection; working fluids.

1. Introduction

The design of cooling systems for the electronic equipment is becoming more viable and challenging, as the demand for more reliable and faster electronic systems has increased rapidly. It is evident that overheating of electronic equipment reduces the overall performance of the electronic systems. In order to maintain their component temperature at the desired limit (85°C), different cooling methods were employed. In 1975 the computers used were typically huge industrial commercial machines with very low CPU power. Also, the heat dissipated from them was naturally without the use of a fan. With time, the power applied to the electronic chips was increased rapidly; therefore the heat transfer from the chips has also been increased significantly. However, the increased circuit densities and the decrease in the component size have made the thermal control of electronics more complex and difficult to maintain their temperature as low as possible as stated by Peterson and Ortega [1]. Reynell [2] has shown that the temperature is the major cause for the failure of electronic equipment. The integrated chips, which are mounted on the printed circuit board, are made of silicon, which is very sensitive to temperature.

Air cooling techniques can be used for low-power consumer electronic modules, like personal computers, notebooks, CPU and GPU of smaller sizes. However, water cooling is a robust cooling method used for cooling of IC engines, large industrial systems like steam power plants, generators, machine gun barrel and lubricant oils in a pump, which involves high heat dissipation rates. However, the high-end electronic devices are cooled using FC-72 involving boiling and condensation. Lasance and Simons [3] suggested the heat flux ranges for different fluids and are given as follows: air, 50 W/cm^2; FC-72, 5–55 W/cm^2 and water, 500–1000 W/cm^2.

The free, forced and mixed convection cooling methods can be identified and used on the basis of the type of the electronic components required to be cooled, their size, their heat dissipation rate and type of fluid used to cool the components. Free convection cooling is an ideal way for low-power electronic components having heat dissipation rate (q) less than 50 W/cm^2 ($h \sim 0.1$ W/cm^{2}K) as suggested by Lasance and Simons [3]. However, for high-power electronic components, forced convection cooling can be applied having heat dissipation rate of 300–1000 W/cm^2 ($h \sim 10$ W/cm^{2}K) as suggested by Overholt *et al* [4]. However, mixed convection draws a balance between both free and forced convection, and has a huge application for cooling of electronic components, and can

*For correspondence

be used for a heat dissipation rate of 20–200 W/cm^2 as mentioned by Mahaney *et al* [5].

2. Background study

Numerous works have been carried out for the cooling of electronic components using different working fluids like air, water, fluorocarbon liquids (FC-72), dielectric fluids (R-113, HFE-7200) and nano-fluids (Al_2O_3, CuO, TiO_2, etc.). The choice of the working fluids depends on their boiling point, which helps for a better rate of heat removal from the electronic components and keeps their cooling level at optimal. The boiling point of the fluid plays an important role in the heat removal from the electronic components. By increasing the boiling point (saturation temperature, t_{sat}) of fluids, their saturation pressure (P_{sat}) increases. This leads to reduction in the length of liquid–vapour line (enthalpy of vaporization, h_{fg} line) of the fluid, which is well known from the phase transformation diagram of pure substance. Hence, the liquid will take less time to be converted into vapour and vice versa. This confirms that use of high-boiling-point fluid leads to better heat removal from the electronic components keeping other parameters fixed (input heat flux, input velocity, size and position of components). Hence, the high-boiling-point fluid can accommodate the spatial and temporal power variations, minimize the electronic component temperature and control their failure rates.

Experimental studies for the cooling of protruding discrete heat sources mounted at different positions on a substrate board using air as the working medium was carried out by Hotta *et al* [6]. The study confirms the effect of surface radiation, which reduces the temperature of electronic components by as much as 12%. Cooling of high-power electronic equipment with arrays of micro-jets using deionized water and FC-40 was carried out by Fabbri and Dhir [7] and they achieved a maximum surface heat flux of 310 W/cm^2. They found that convective heat transfer is enhanced by increasing the Reynolds number and Prandtl number. Numerical investigation of two-phase jet impingement cooling of a square electronic chip (silicon) using FC-72 and a dielectric liquid was carried out by Wang *et al* [8], who found that the two phase cooling results in 10% decrease in temperature of the chip as compared with the single-phase cooling. Analytical and experimental studies for natural and forced convective heat transfer from small electronic devices were carried out by Baker [9]. They predicted the average convective heat transfer coefficient of electronic devices, which increased significantly on decreasing the size of the devices. Experimental investigations of geometric and flow parameters for high-power electronic components on their heat transfer characteristics using unsteady and steady jets were carried out by Leena *et al* [10]. They found that the enhancement is due to the construction and destruction of thermal and hydrodynamic boundary layer formation. Experimental investigation of natural convection heat transfer enhancement from a 3 × 3 square (12.7 mm × 12.7 mm) array of IC chips was carried out by Heindel *et al* [11], who found that the heat transfer is enhanced by 24 times in the vertical orientation and by 15 times in the horizontal orientation of heat sources. Cooling of 12 flush mounted square discrete heat sources (12.7 mm × 12.7 mm) mounted on a horizontal rectangular channel filled with FC-77 and water was carried out by Incropera *et al* [12]. They found that the average convective heat transfer coefficient for the rows of the array was decreased by 25% from the first to the second row, and 5% from the third to the fourth row. Experimental investigation of steady and transient heat transfer from eight in-line rectangular heated protrusions mounted on a vertical channel under natural convection was reported by Joshi *et al* [13]. They used water as a coolant and concluded that, for smaller spacing, the component surface temperature increased significantly due to the reduction in fluid velocities. Experimental analysis for cooling of a microprocessor board using water as a cooling medium was carried out by Sharma *et al* [14]. They have observed the heat flux removal and temperature drop of around 2–3 W/cm^2 and 65°C, respectively. Experimental investigation has been carried out by Bhowmik and Tou [15] on square discrete heat sources mounted on a vertical channel filled with FC-72 and water, which suggested that cooling was more effective using FC-72. They have also proposed a correlation among the Nusselt number, Peclet number and Fourier number of fluid.

It has been confirmed from the literature that most of the studies [6, 10, 11, 13, 14] are pertinent to the use of single working fluid under single mode of heat transfer for the cooling of discrete heated modules. Some of the researchers [7–9, 12, 15, 16] have also focused on the use of two working fluids. However, the present work emphasizes on the role of three different working fluids (air, water and FC-72) under three different modes of heat transfer (free, forced and mixed convection) for the cooling of discrete heated modules. This will give a clear insight for the electronic cooling engineers regarding the selection of working fluids and modes of heat transfer for the cooling of electronic components. Hence, the study is critical for the thermal management designers. Again, most of the studies were focused on identical heat sources. The analysis of non-identical heat sources is scarce. Hence the present work is focused on cooling of non-identical heat sources using different working fluids under different modes of heat transfer.

3. Methodology

Numerical analysis is carried out using ANSYS-Icepak for the cooling of three electronic chips mounted on a substrate board. The analysis is carried out using three different

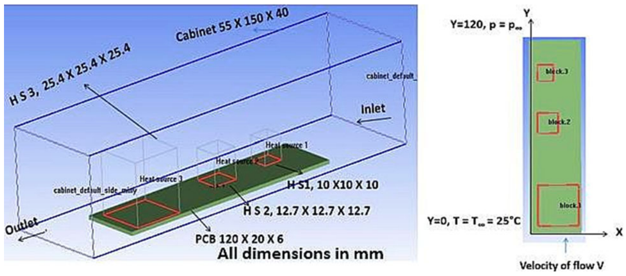

Figure 1. Computational model used for the present study.

working fluids (air, FC-72 and water) under all the three modes of heat transfer (free, forced and mixed convection).

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 FLUENT is used to solve the governing equations, such as the conservation equations of mass, momentum, energy and other scalars such as turbulence if needed. The FLUENT solver is based on the control volume technique. The model used in the present study consists of a test section (Cabinet), the substrate (Bakelite) board and three non-identical heat sources (Aluminium) mounted on the substrate board [15], as shown in figure 1.

3.1 *Optimality test using golden mean ratio (GMR)*

The effect of the golden mean ratio (GMR) is analysed in the present study for a particular configuration under forced convection ($V = 4$ m/s) using water as the working fluid and is shown in figure 2.

The GMR is an important geometrical parameter for the arrangement of heat sources of a configuration that reduces the maximum temperature of the configuration by as much as 10%, as reported by Liu and Phan-Thien [17]. This is basically arranging the heat sources of a configuration where the distance between these follow a geometric ratio (GMR) of 1.618 instead of equidistance arrangement. The results suggest that there is a significant temperature drop of 2–3°C (5%) by arranging the heat sources following GMR, which controls the interaction effect between the heat sources and leads to better cooling.

3.2 *Governing equations*

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

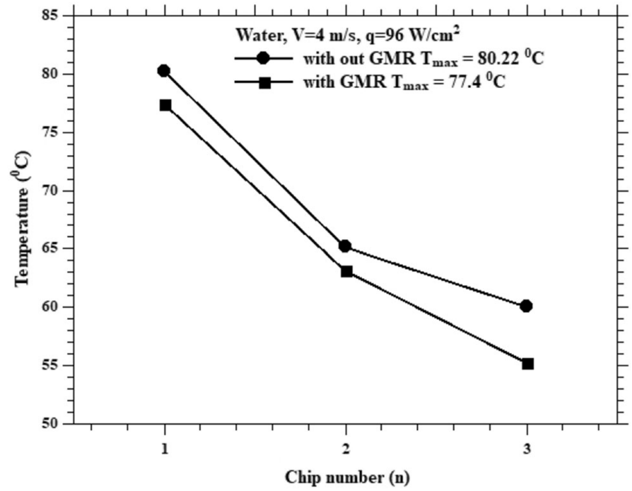

Figure 2. Effect of GMR on the temperature of different chips.

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

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

For an incompressible fluid, where the value of density is constant, Eq. (1) reduces to the form of Eq. (2):

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

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 the steady flow is given in Eqs. (3)–(5):

$$u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} + w\frac{\partial u}{\partial z} = -\frac{1}{\rho}\frac{\partial P}{\partial x} + \nu\left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2}\right), \tag{3}$$

$$u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y} + w\frac{\partial v}{\partial z} = -\frac{1}{\rho}\frac{\partial P}{\partial y} + \nu\left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial z^2}\right) + g\beta(T - T_\infty), \tag{4}$$

$$u\frac{\partial w}{\partial x} + v\frac{\partial w}{\partial y} + w\frac{\partial w}{\partial z} = -\frac{1}{\rho}\frac{\partial P}{\partial z} + \nu\left(\frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2} + \frac{\partial^2 w}{\partial z^2}\right). \tag{5}$$

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):

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

Table 1. Heat flux values (W/cm^2) for different working fluids under different modes of heat transfer.

Working fluid (→)			
Modes of heat transfer (↓)	Air	FC-71	Water
Natural convection	0.38	0.5	2.75
Forced convection ($V = 2$ m/s)	1.8	34	80
Forced convection ($V = 4$ m/s)	2.1	52	95
Forced convection ($V = 6$ m/s)	2.4	62	102
Mixed convection	0.75	2.1	2.96

where q_g is the heat generation, C is the specific heat and Φ is the viscous dissipation, which is further written in the form as given in Eq. (7):

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

3.3 *Boundary conditions*

The boundary conditions for the natural, forced and mixed convection are given in Eqs. (8) and (9). The board is vertical; hence the working fluids flow in a vertical direction against the gravity. The boundary conditions for the natural convection, forced convection (V = 2, 4 and 6 m/s) for all the working fluids (air, water, FC-72) and mixed convection (air, V = 2.25 m/s; water, V = 0.94 m/s; FC-72, V = 2.74 m/s) are given as follows. The mixed convection velocity is calculated in such a way that the maximum temperature of the heat sources should not exceed 85°C. Suitable values of heat flux were supplied to the heat sources using different working fluids to determine their mixed convection velocity. The calculation for the same is explained in section 4.3.

$$\begin{aligned} &At\ Y = 0, T = T_\infty = 25°\mathrm{C}, \\ &\quad V(only\ for\ forced\ and\ mixed\ convection) \end{aligned} \tag{8}$$

$$At\ Y = 120\ \mathrm{mm}, p = p_\infty. \tag{9}$$

The heat flux supplied to the bottom of the heat sources for different working fluids is given in table 1. The specifications of different heat sources (Aluminium) and the substrate board (Bakelite) used for the present study are given in table 2.

3.4 *Grid independence study*

The ANSYS-Icepak uses FLUENT solver, which is based on control volume technique (implicit scheme). The pre-processing generated a HEXA-dominant mesh, which gives better results and has all the components of standard geometries.

Table 2. Specification of different components (in mm) for the present study [15].

Components	Length L	Width W	Thickness t
Substrate	120	20	6
Heat source 1	10	10	10
Heat source 2	12.7	12.7	12.7
Heat source 3	25.4	25.4	25.4

In the present work, three fluids are used; hence three different mesh elements will be considered. The mesh profile used for the simulation of forced convection medium with water as the working fluid is shown in figure 3. The figure gives the mesh profile for water with a ±0.5°C variation of temperature under forced convection medium. Tables 3 and 4 give the complete mesh profile for different working fluids under natural and mixed convection heat transfer mode. The calculations are based on grid convergence index (GCI) as given by Roache [18]. A similar analysis has also been used for forced convection mode of heat transfer. The optimal grid sizes are highlighted in the table.

4. Results and discussion

A numerical analysis is carried out to study the heat transfer characteristic of three heat sources arranged on a substrate board using GMR. The analysis is carried out using three working fluids (air, FC-72 and water) under different modes of heat transfer.

4.1 *Natural convection*

The numerical investigation is carried out for three non-identical heat sources using different working fluids under

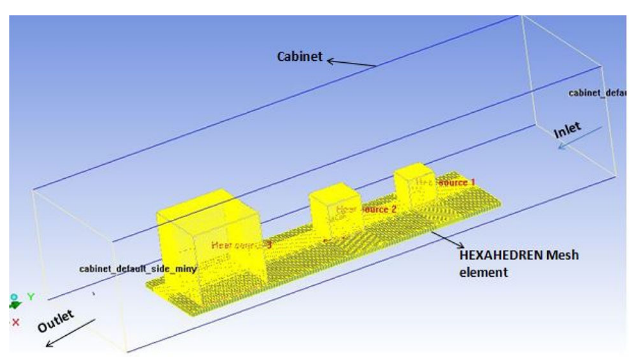

Figure 3. Mesh profile used for simulation of forced convection using water.

Table 3. Mesh specification for three working fluids under natural convection heat transfer.

Working fluids	NODES	HEAXS	QUADS	T_{max}	Continuity equation	Energy equation	Momentum equation X	Momentum equation Y	Momentum equation Z
Air	1696500	1662351	97301	85.81	5.8×10^{-4}	8.32×10^{-6}	8.01×10^{-3}	2.043×10^{-3}	$7.01 \times 10-3$
	1871748	**182548**	**104623**	**84.6**	$\mathbf{5.026 \times 10^{-5}}$	$\mathbf{9.11 \times 10^{-8}}$	$\mathbf{8.37 \times 10^{-4}}$	$\mathbf{2.05 \times 10^{-4}}$	$\mathbf{7.52 \times 10^{-4}}$
	1970128	1898153	91803	86.2	4.12×10^{-6}	9.632×10^{-9}	5.21×10^{-5}	1.67×10^{-5}	5.98×10^{-5}
FC-72	1696500	1662351	97301	77.29	5.8×10^{-4}	8.32×10^{-6}	8.01×10^{-3}	2.043×10^{-3}	7.01×10^{-3}
	1871748	**182548**	**104623**	**81.2**	$\mathbf{5.026 \times 10^{-5}}$	$\mathbf{9.11 \times 10^{-8}}$	$\mathbf{8.37 \times 10^{-4}}$	$\mathbf{2.05 \times 10^{-4}}$	$\mathbf{7.52 \times 10^{-4}}$
	1970128	1898153	91803	88.73	4.12×10^{-6}	9.632×10^{-9}	5.21×10^{-5}	1.67×10^{-5}	5.98×10^{-5}
Water	1696500	1662351	97301	74.23	5.8×10^{-4}	8.32×10^{-6}	8.01×10^{-3}	2.043×10^{-3}	7.01×10^{-3}
	1871748	**182548**	**104623**	**79.6**	$\mathbf{5.026 \times 10^{-5}}$	$\mathbf{9.11 \times 10^{-8}}$	$\mathbf{8.37 \times 10^{-4}}$	$\mathbf{2.05 \times 10^{-4}}$	$\mathbf{7.52 \times 10^{-4}}$
	1970128	1898153	91803	86.01	4.12×10^{-6}	9.632×10^{-9}	5.21×10^{-5}	1.67×10^{-5}	5.98×10^{-3}

Table 4. Mesh specification for three working fluids under mixed convection heat transfer.

Working fluids	NODES	HEAXS	QUADS	T_{max}	Continuity equation	Energy Equation	Momentum equation X	Momentum equation Y	Momentum equation Z
Air	1286351	1271503	80102	79	1.98×10^{-4}	5.81×10^{-7}	5.2×10^{-3}	1.59×10^{-3}	5.1×10^{-3}
	1347108	**1338600**	**82192**	**82.3**	$\mathbf{3.383 \times 10^{-5}}$	$\mathbf{7.78 \times 10^{-8}}$	$\mathbf{7.12 \times 10^{-4}}$	$\mathbf{1.88 \times 10^{-4}}$	$\mathbf{6.57 \times 10^{-4}}$
	1437219	1429000	95108	83.2	3.811×10^{-6}	7.32×10^{-9}	6.94×10^{-4}	1.81×10^{-5}	6.01×10^{-5}
FC-72	1286351	1271503	80102	78.8	1.98×10^{-4}	5.81×10^{-7}	5.2×10^{-3}	1.59×10^{-3}	5.1×10^{-3}
	1347108	**1338600**	**82192**	**80.1**	$\mathbf{3.383 \times 10^{-5}}$	$\mathbf{7.78 \times 10^{-8}}$	$\mathbf{7.12 \times 10^{-4}}$	$\mathbf{1.88 \times 10^{-4}}$	$\mathbf{6.57 \times 10^{-4}}$
	1437219	1429000	95108	81.6	3.811×10^{-6}	7.32×10^{-9}	6.94×10^{-4}	1.81×10^{-5}	6.01×10^{-5}
Water	1286351	1271503	80102	78	1.98×10^{-4}	5.81×10^{-7}	5.2×10^{-3}	1.59×10^{-3}	5.1×10^{-3}
	1347108	**1338600**	**82192**	**78.4**	$\mathbf{3.383 \times 10^{-5}}$	$\mathbf{7.78 \times 10^{-8}}$	$\mathbf{7.12 \times 10^{-4}}$	$\mathbf{1.88 \times 10^{-4}}$	$\mathbf{6.57 \times 10^{-4}}$
	1437219	1429000	95108	79.3	3.811×10^{-6}	7.32×10^{-9}	6.94×10^{-4}	1.81×10^{-5}	6.01×10^{-5}

natural convection mode of heat transfer. Natural convection is the most common and simplest technique employed for cooling of low-power electronic devices.

4.1a *Variation of maximum temperature for different working fluids under natural convection*: It is clear from figure 4 that the choice of working fluids depends on their boiling point (air: − 193.2°C, FC-72: 56.4°C, water: 100°C), i.e, the fluid with high boiling point will lead to greater heat removal rate from the electronic components and hence helps in keeping their cooling level at optimal. The explanation for the same is given in section 2. Due to the high boiling point of water, the maximum temperature among the three heat sources of the configuration is the lowest for water. This will lead to higher heat removal from the heat sources; hence, cooling will be predominant using water as the working fluid. Hence, water can be used as the ideal fluid for cooling of high-power electronic components.

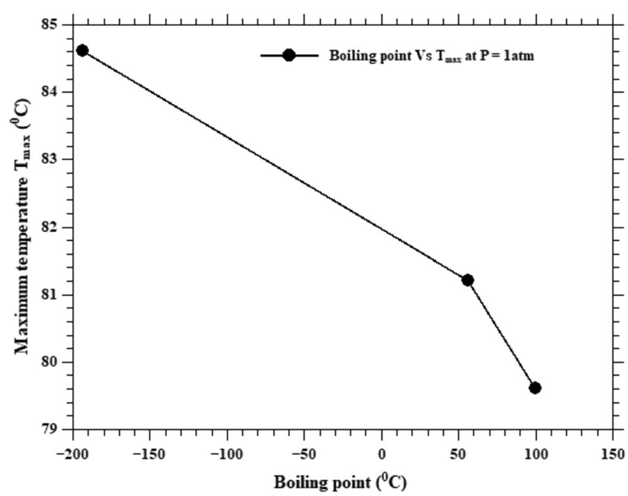

Figure 4. Variation of maximum temperature of the configuration using different working fluids under natural convection.

4.1b *Temperature variation of different heat sources under natural convection*: It is seen from figure 5 that the temperature of the heat sources is the lowest using water as the working fluid, due to its high boiling point. Among the three heat sources of the configuration, the temperature increases from heat source 1 to 3. As heat source 3 has the largest size, the heat accumulation for this heat source will be maximum leading to the highest temperature. Hence, it is evident that size of the heat sources plays a significant role for their cooling. Although heat source 3 is placed at

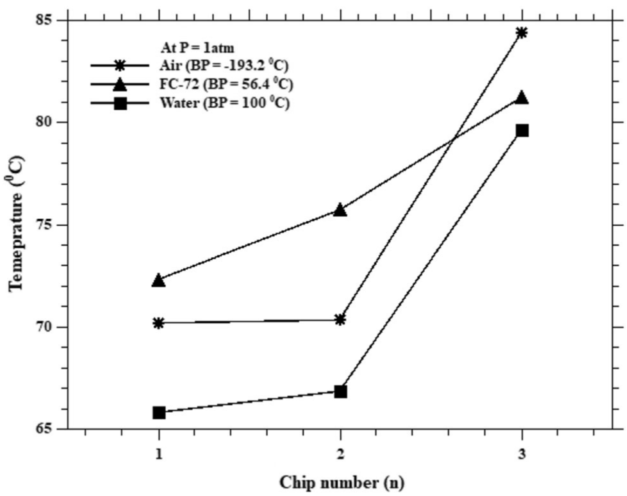

Figure 5. Temperature variation of different heat sources of a configuration under natural convection.

the substrate bottom, due to its largest size, this will lead to the highest temperature.

Figure 6 shows the variation of the Nusselt number of different heat sources using different working fluids. Nusselt number is the dominant non-dimensional parameter responsible for the cooling of discrete heated modules, which ultimately depends on the heat transfer coefficient (h), which is a strong function of the heat dissipation rate and temperature of heat sources.

It is clear from the figure that cooling will be more effective using water as the working fluid. This is due to the fact that water reduces the temperature of electronic components significantly due to its high boiling point value, which will lead to a better cooling of components. Among the three chips of a particular configuration, chip 1 having smaller size will be cooled at a faster rate, due to the dominant velocity and thermal boundary layer formation. This clearly indicates the role of size of the heat sources for their cooling.

4.2 *Forced convection*

The requirement of heat removal from high-power electronic devices makes the forced convection cooling indispensable. Hence forced convection becomes a convenient mode for cooling of high-power electronic components. The numerical simulations are carried out for cooling of heat sources under forced convection medium using three different velocities (2, 4 and 6 m/s), and with different working fluids.

4.2a *Variation of maximum temperature for different working fluids under forced convection*: Heat transfer studies are performed at three different velocities using three working fluids, under forced convection medium, as shown in figure 7. It is clear from the figure that water has higher heat removal rate as compared to air and FC-72, because of its high boiling point value. Hence, water plays an important role in cooling of discrete heated modules. On increasing the velocity of working fluid from 2 to 6 m/s, the cooling becomes much faster, and temperature of heat sources drops by 2–3°C. Hence, the working fluids along with their velocity must be selected carefully to keep the temperature level of heat sources at optimal.

4.2b *Variation of the temperature of different heat sources for different working fluids under forced convection at $V = 6$ m/s*: The numerical simulations were carried out for three heat sources to study their temperature variation using different working fluids at $V = 6$ m/s. It is clear from figure 8 that heat source 1 has the lowest temperature due to its smallest size, and the temperature increases gradually from the heat source 2 to 3. Again, due to the high boiling point, the temperature of the heat sources using water is the

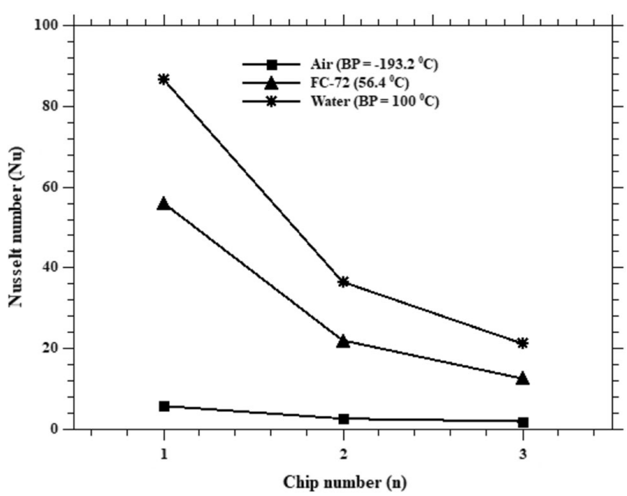

Figure 6. Variation of Nusselt number of different heat sources of a configuration under natural convection.

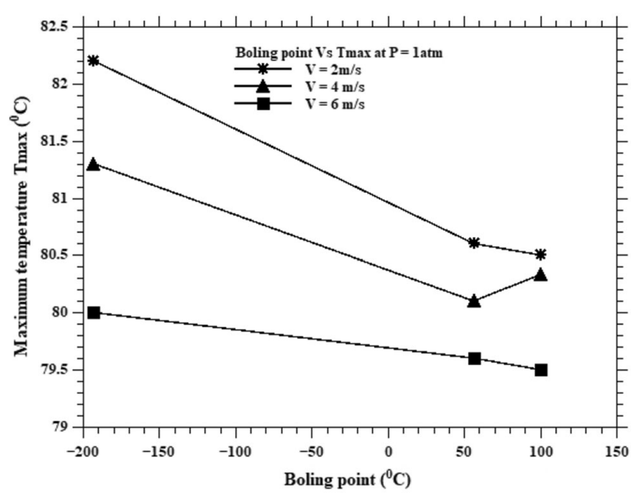

Figure 7. Variation of maximum temperature of the configuration using different working fluids under forced convection.

lowest. Due to the smallest size, the formation of velocity and thermal boundary layer will be much faster, and this will lead to a better cooling of heat sources. Hence, size of the heat sources is significant for their cooling along with the heat dissipation rate and input velocity of the fluid.

4.3 *Mixed convection*

Mixed convection mode of heat transfer has the advantage of drawing a balance between both free and forced convection modes. Hence this is widely employed in the electronics industry for the cooling of discrete heated modules. The following formulation has been considered to obtain the velocities of working fluids under mixed convection medium. The mixed velocity is calculated keeping in mind the safe temperature limit of heat sources, i.e., 85°C. A suitable value of heat flux is supplied to get the required velocity to maintain this safe temperature.

Consider FC-72 as the working fluid, whose necessary parameters are given here.

$$L \text{ (maximum length among three heat sources)} = 25.4 \text{ mm},$$
$$\Delta T = T_{max} - T_{\infty} = 85°\text{C} - 25°\text{C} = 60°\text{C},$$
$$k_{T_{mean}} = k_{55°\text{C}} = 0.05395 \text{ W/m [19]},$$
$$q = 2.1 \text{W/cm}^2,$$
$$1 \simeq \frac{Gr}{Re^2},$$
$$1 \simeq \frac{g\beta\Delta T L^3/v^2}{(VL/v)^2},$$
$$V^2 = g\beta\Delta T L,$$
$$\beta = \frac{1}{T_{mean}} = \frac{1}{55+273} = 0.00305\text{K}^{-1},$$
$$q = k\frac{\Delta T}{\Delta x},$$
$$\Delta T = \frac{qL}{k},$$
$$V^2 = \frac{g\beta q L^2}{k},$$
$$V^2 = \frac{9.81 \times 0.00305 \times 2.1 \times 10^4 \times (25.4 \times 10^{-4})^2}{0.05395},$$
$$V \simeq 2.74 \text{ m/s}.$$

4.3a *Variation of maximum temperature of the configuration using different working fluids under mixed convection*: Figure 9 shows the variation of maximum temperature among different heat sources of the configuration using different working fluids under mixed convection mode. To achieve the mixed convection range, the heat sources are supplied with different working fluids having different fluid velocities (section 3.2). Among all the heat sources, the maximum temperature is obtained for heat source 3, even if it is supplied with high velocity. The maximum temperature among the heat sources of a configuration is

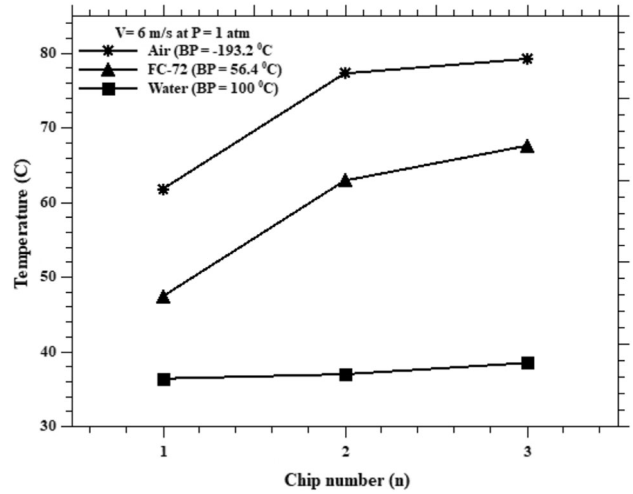

Figure 8. Temperature variation of different heat sources in forced convection at velocity $V = 6$ m/s.

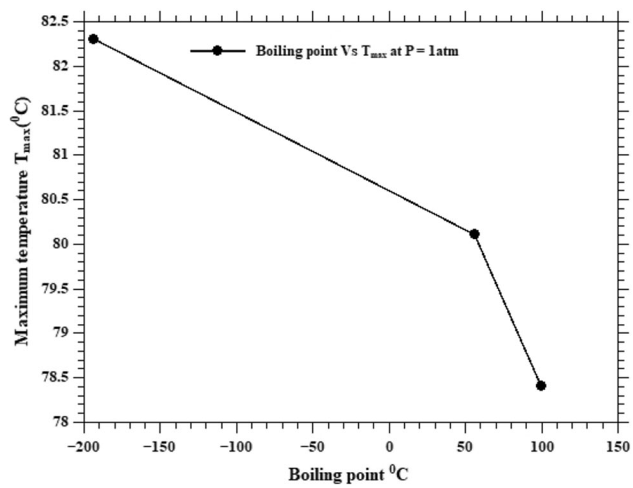

Figure 9. Variation of maximum temperature of the configuration using different working fluids in mixed convection.

the lowest for water; hence, this can be used for better heat removal from the electronic components under mixed convection.

4.3b *Effect of different modes of heat transfer on the temperature of different heat sources using different working fluids*: Figure 10 gives a comparison between the different heat transfer modes along with different working fluids used for the cooling of discrete heated modules. The natural convection heat transfer has the lowest heat removal rate and forced convection has the highest heat removal rate as this is achieved by maintaining some velocity. However, the mixed convection mode of heat transfer draws a balance between free and forced convection modes and is able to maintain the optimal temperature of heat sources on the substrate. Hence the heat removal from the electronic components will be in the following order:

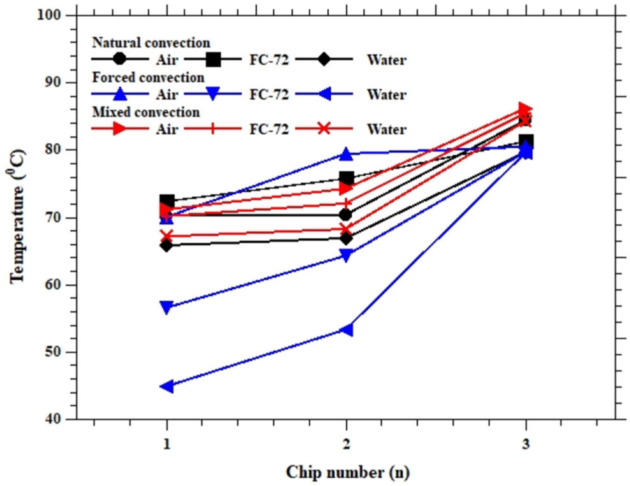

Figure 10. Effect of different modes of heat transfer and different types of working fluids on cooling of heat sources.

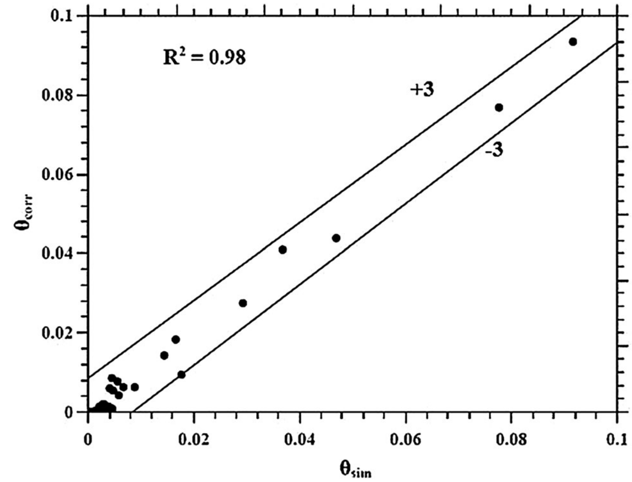

Figure 11. Parity plot showing the agreement between non-dimensional simulated temperature (θ_{sim}) and non-dimensional correlated temperature (θ_{corr}).

- forced convection water (boiling point BP = 100°C)
- forced convection FC-72 (BP = 56.4°C)
- forced convection air (BP = –193.2°C)
- mixed convection water
- mixed convection FC-72
- mixed convection air
- natural convection water
- natural convection FC-72 and
- natural convection air

4.4 *Proposed correlation*

To generalize the discussion and for the reference of wide heat transfer community, a correlation is proposed for the maximum temperature of the heat sources. It is evident from the discussion that the maximum temperature of the heat sources is a strong function of their size (X), position (Y), type of working fluid (boiling point θ_b), the velocity of fluid flow (V) and heat flux (q^*). However, it is clear from the whole discussion in the manuscript that the size of the heat sources and their positions on the board are fixed. Hence, a correlation is proposed for the non-dimensional maximum temperature in terms of non-dimensional boiling point of working fluid, non-dimensional velocity of fluid flow and non-dimensional heat flux values. The correlation is based on 48 data points with a regression coefficient (R^2) of 0.98 and RMS error of 3%. The correlation is given in Eq. (10):

$$\theta = (1+\theta_b)^{-1.725}(1+V^*)^2(q^*)^{-1.73}. \qquad (10)$$

Equation 10 is valid for the following range of parameters:

$$0.21 \le \theta_b \le 1,$$
$$0.0018 \le V^* \le 1,$$
$$2.57 \le q^* \le 924.$$

Based on this correlation, a parity plot is plotted between the simulated and correlated values of the non-dimensional temperature and is shown in figure 11. It is clear from the figure 11 that both values agree well with each other within an error band of $\pm 3\%$. Thus, the correlation can predict the whole physics embedded in the problem under consideration significantly. Hence, this will help the electronic cooling engineers to predict the component temperatures accurately without carrying out further numerical simulations.

5. Conclusions

Three dimensional steady-state numerical simulations are carried out using ANSYS-Icepak for the three protruding discrete heat sources arranged on a substrate board following GMR. The simulations are carried out under different modes of heat transfer (free, forced and mixed) using different working fluids (water, FC-72 and air). It is seen that the boiling point of working fluids plays an important role in the heat removal from the heat sources. The heat removal from the heat sources is the highest using water, due to its high boiling point value, under all the modes of heat transfer. Again the size of the heat sources also plays a significant role in their cooling. It is evident that heat source 3 (largest size), although placed at the bottom of the substrate board, is subjected to the highest temperature. However, the cooling is much faster for the smallest size heat sources due to the faster development of velocity and thermal boundary layers. Hence it confirms that the temperature of the heat sources is a strong function of their size, position on the substrate board, the velocity of the fluid and type of working fluid used.

Nomenclature

g	acceleration due to gravity, 9.81 m/s^2
Gr	Grashof number, $\frac{g\beta\Delta TL^3}{\nu^2}$
h	heat transfer coefficient, W/m^2 K
k	thermal conductivity, W/m K
L	length of the heat source, m
q	heat flux, W/m^2
q^*	non-dimensional heat flux, $\frac{\Delta T_{ref}}{T_\infty}$
Q	heat supplied, W
Re	Reynolds number, $\frac{VL}{\nu^2}$
Ri	Richardson number, $\frac{Gr}{Re^2}$
T	temperature, K
V	velocity of fluid, m/s
V^*	non-dimensional velocity, $\frac{V}{V_{max}}$

Greek symbols

β	isobaric thermal expansion coefficient of fluid, $\frac{1}{T_{mean}}$, K^{-1}
ΔT_{ref}	reference temperature, $\frac{qL}{k_f}$, K
ν	kinematic viscosity of fluid, m^2/s
θ	non-dimensional temperature, $\frac{T_{max} - T_\infty}{\Delta T_{ref}}$

Subscripts

∞	ambient
f	fluid
max	maximum

References

[1] Peterson G P and Ortega A 1990 Thermal control of electronic equipment and devices. *Adv. Heat Transf.* 20: 181–314

[2] Reynell M 1990 *Major causes for failure of electronic components*. U.S. Air Force Avionics Integrity Program

[3] Lasance C J and Simons R E 2005 Advances in high-performance cooling for electronics. *Electron. Cool. Mag.* 11: 22–39

[4] Overholt M R, McCandless A, Kelly K W, Becnel C and Motakef S 2005 Micro-jet arrays for cooling of electronic equipment. In: *Proceedings of the* 3rd *ASME International Conference on Micro-channels and Mini-channels*, pp. 249–252

[5] Mahaney H V, Incropera F P and Ramadhyani S 2007 Measurement of mixed-convection heat transfer from an array of discrete sources in a horizontal rectangular channel with and without surface augmentation. *Exp. Heat Transf.* 3: 215–237

[6] Hotta T K, Muvvala P and Venkateshan S P 2013 Effect of surface radiation heat transfer on the optimal distribution of discrete heat sources under natural convection. *Heat Mass Transf.* 49: 207–217

[7] Fabbri M and Dhir V K 2005 Optimized heat transfer for high power electronic cooling using arrays of the microjet. *ASME J. Heat Transf.* 127: 760–769

[8] Wang D, Yu E and Pezekwas A 1990 A computational study of two-phase jet impingement cooling of an electronic chip. In: *Proceedings of the* 15th *Annual IEEE Semiconductor Thermal Measurement and Management Symposium*, pp. 10–15

[9] Baker E 1972 Liquid cooling of microelectronic devices by free and forced convection microelectronic and reliability. *Microelectron. Reliab.* 11: 213–222

[10] Leena R, Renjith R V and Prakash M J 2015 Experimental and numerical investigations on steady and unsteady jet impingement cooling for high power electronics. *IEEE Trans. Components, Packag. Manuf. Technol.* 5: 636–640

[11] Heindel T J, Ramadhyani S and Incropera F R 1995 Conjugate natural convection from an array of discrete heat sources part-I – two and three dimensional model validations. *Int. J. Heat Mass Transf.* 16: 501–510

[12] Incropera F P, Kerby J S, Moffatt D F and Ramadhyani S 1986 Convective heat transfer from discrete heat sources in a rectangular channel. *Int. J. Heat Mass Transf.* 29: 1051–1058

[13] Joshi Y, Wilson T and Hazard S J 1989 An experimental study of natural convection cooling of an array of heated protrusions in a vertical channel in water. *ASME J. Electron. Packag.* 111: 33–40

[14] Sharma C S, Zimmermann S, Tiwari M K, Michel B and Poulikakos D 2012 Optimal thermal operation of liquid-cooled electronic chips. *Int. J. Heat Mass Transf.* 55: 1957–1969

[15] Bhowmik H and Tou K W 2005 Study of transient forced convection heat transfer from discrete heat sources in an FC-72 cooled vertical channel. *Int. J. Therm. Sci.* 44: 499–505

[16] Park K A and Bergles A E 1987 Natural convection heat transfer characteristics of simulated microelectronic chips. *ASME J. Electron. Packag.* 109: 90–96

[17] Liu Y and Phan-Thien N 2000 An optimum spacing problem for three chips mounted on a vertical substrate in an enclosure. *Numer. Heat Transf. Part A Appl.* 37: 613–630

[18] Roache P J 1994 Perspective: a method for uniform reporting of grid refinement studies. *J. Fluid Eng.* 116: 405–413

[19] *fluorinert-electronic-liquid-fc-72.pdf*