Natural Convection from Discrete Heat Sources in Enclosures: An Overview

G.S.V.L. Narasimham*
Department of Mechanical Engineering, Indian Institute of Science, Bangalore
*Corresponding author e-mail: mecgsvln@mecheng.iisc.ernet.in

Abstract
Natural convection is an inexpensive, highly dependable, vibration- and noise- free mechanism for heat removal from electronic packages and can cope up with heat fluxes of the order of 10^3 $W.m^{-2}$ without the package temperatures exceeding typically 80-90 ^0C. In order to increase the natural convection heat removal rates, some techniques like liquid immersion cooling with heat sink enhancement are being tried. In this overview, the work done in the field of natural convection from discrete heat sources in enclosures is reviewed and directions for further research are mentioned.

Keywords: Natural Convection, Heat Removal, Discrete Heat Source.

Nomenclature

a	Thermal diffusivity	$m.s^{-2}$
g	Acceleration due to gravity	m/s^2
Gr	Grashof number $[(g\beta\Delta T L_c^3)/(\upsilon_f a_f)]$	Dimensionless
H	Height of the enclosure	m
L	Breadth of the enclosure (fluid gap)	m
L_c	Characteristic dimension(heater height or enclosure height)	m
N	Number of heaters	Dimensionless
Nu	Nusselt number $(\alpha L_c / \lambda_f)$	Dimensionless
Pr	Prandtl number (υ_f / a_f)	Dimensionless
$q^{\cdot}$	Heat flux	$W.m^{-2}$
Q	Power dissipation	W
Q_L	Linear heat generation rate	$W.m^{-1}$
Q_υ	Volumetric heat generation rate	$W.m^{-3}$
Ra	Rayleigh number (Gr. Pr)	Dimensionless
T	Temperature	^0C
W	Width of the enclosure	m

(i.e., dimension perpendicular to the cross-section of the main fluid circulation

Greek letters

α	Heat transfer coefficient	$W.m^{-2}.^0C^{-1}$
β	Thermal volumetric expansion coefficient	K^{-1}
ΔT	Characteristic temperature difference $((T_h - T_c)$, qLc/λ_f , $^{\cdot}Q_L/\lambda_f$, $^{\cdot}Q_\upsilon L_c^2/\lambda_f$	^0C
η	Dynamic viscosity	Pa.s
λ	Thermal conductivity	$W.m^{-1}. ^0C^{-1}$
υ	Kinematic viscosity (η/ρ)	$m^2.s^{-1}$
ρ	Density	kg/m^3

Subscripts
c - cold; f - fluid; h - hot; max - maximum; p - protrusion, heater, block; s - substrate;
Superscript
* - Dimensionless quantity

1. Introduction

Natural convection encompasses such situations where the fluid motion arises due to density differences in the presence of force fields like gravitational and centrifugal. The density differences may be due to temperature and/or concentration variations. Gebhart (1971) points out that one body of usage terms the external natural convection as free convection, and the internal natural convection, as natural convection. Another usage calls both the internal and external cases the free convection. However, the term "buoyancy-driven convection" may be more appropriate since it gives an idea of the actual mechanism. Natural convection problems are substantially more difficult to investigate than forced convection problems due to the coupling between the flow and heat transfer, but they occur very frequently in a number of applications.

In the present overview, attention is focused on natural convection arising in enclosures due to discrete heat sources. Such problems have practical application in the area of the cooling of electronic packages and equipment. The results may also have application in room heating because similar problems also arise in case of rooms heated by baseboard-type heaters attached to one of the vertical walls of the room.

Natural convection is an inexpensive, noise-free and reliable means of heat transport since it does not depend upon the functioning of external agents like pumps and blowers.

Since the development of the first electronic digital computers, heat removal has played an important role in maintaining reliable operation in the electronic equipment. From the large-scale integration technologies to very large-scale technologies there has been a steady increase in the heat dissipation at chip, module and system level. Today, the heat flux level of high-performance chips is typically 5 × 105 W.m^{-2} and this figure is expected to exceed 106 W.m^{-2} shortly in this century. For the heat removal of this magnitude, special and very complex cooling technologies like direct air forced convection, liquid immersion cooling, high conduction coupled liquid cooling, microchannel cooling and immersion-boiling have been developed.

However, in a number of other applications, where the heat flux levels are typically of the order of 103 W.m^{-2} and where the upper limit on the operating temperature is typically 100^{0}C, natural convection is still the preferred method for the cooling of a variety of electronic equipment in view of its simplicity and reliability. Examples of such applications include communication switching devices, avionics packages, electronic test equipment, consumer electronics and low-end computer packages. These devices encompass a variety of geometrical configurations like vertical and inclined parallel plate channels, vented enclosures and completely sealed enclosures and the like. In view of the reliability of natural convection, various schemes involving liquid immersion cooling along with heat transfer enhancement devices for are being investigated in order to raise the natural convection heat flux removal levels as much as possible. The applicability of various cooling techniques is shown in Fig. 1, while Fig. 2 presents the technology trends in the transistor and chip power densities.

Heat transfer from electronic equipment generally occurs through all the three modes mentioned earlier. In most of the cases, conduction is the primary mode of heat transfer from the point of heat generation. By this mode, heat transfer takes place from the junction of the integrated circuit to the case and then to the circuit board through the pins. Convection and radiation are the main means of dissipation of heat from the circuit board to the cooling medium, which is the ultimate heat sink for the package.

There exist a number of review papers on the subject of the thermal management of electronic equipment. Some of the exhaustive reviews with emphasis on heat transfer aspects are: Nakayama (1986, 1996), Incropera (1988), Peterson and Ortega (1990), and Narasimham and Krishna Murthy (1993). In these reviews, a broad coverage was given to various cooling strategies like natural convection air cooling, natural convection liquid immersion cooling, forced air cooling, jet-impingement cooling, micro-channel cooling and immersion boiling.

The present overview, however, is devoted to a more elaborate review of the following selected topics: (i) Flush-mounted thin discrete heat sources in enclosures with air and liquid natural convection cooling, (ii) Discrete heat sources embedded in a substrate forming one vertical wall of the enclosure, and (iii) Protruding discrete heat sources from a substrate forming one vertical wall of the enclosure.

Some enclosure configurations involving discrete heat sources are shown in Fig. 3.

2. Flush-mounted discrete heat sources in enclosures

2.1 Air cooling

Thin, foil-like, electronic components are often mounted on a wall of the enclosure in several applications like sophisticated aircraft and missile systems. If the components are packed on the wall without any gap between them, it amounts to continuous heat generation. On the other hand, if the components are mounted with gaps in between them, only portions of the wall will be active and this results in discrete heating. The enclosure is then said to consist of partially active walls. Similar problems also arise in case of rooms heated by baseboard-type heaters attached to one of the vertical walls of the room.

The geometry that has been considered by several investigators (Chu and Churchill, 1976; Flack and Turner, 1980; Turner and Flack, 1980; Kuhn and Oosthuizen, 1987) is a two-dimensional enclosure of horizontal dimension L and height H, containing a very thin isothermal or isoflux strip heater of height H_p ($< H$), flush-mounted on one of the vertical walls (say, the left vertical wall), the mid-point of the heater being at a distance x_p from the top corner of the wall (or at a distance $x'_p = H - x_p$ from the bottom corner). The top and bottom walls of the enclosure are insulated, while the opposing vertical wall is maintained isothermal at a lower temperature T_c. Alternatively, the opposing wall as well as the top and bottom walls may be maintained at the temperature T_c. The temperature of the heater (average temperature in case of isoflux heater) may be denoted by T_h, while the heat flux from the heater (average heat flux in case of isothermal heater) may be denoted by $q^{\cdot}$. In the definition of the Grashof number, the characteristic temperature difference may be $(T_h - T_c)$ or $q^{\cdot} L_c/\lambda_f$). The characteristic dimension may be the enclosure height H or the heater height H_p.

Computations performed by Chu and Churchill (1976) for an isothermal heater for Prandtl number Pr = 0.7 and $0 \leq Gr \leq 105$ (enclosure height based Grashof number) have led to the following conclusions:

- The circulation (i.e., the maximum absolute dimensionless stream function $|\psi^*|_{max}$, where $\psi^* = \psi / \upsilon$, ψ being the dimensional stream function) in the enclosure passes through a maximum as x_p/H increases from 0.1 to 0.9 for $H_p/H = 0.2$, the maximum occurring at higher values of x_p/H as Gr is increased.
- The Nusselt number Nu ($=\alpha H_p/\lambda_f$, α being the heat transfer coefficient and λ_f being the thermal conductivity of heat transfer medium) passes through a maximum when x_p/H is increased from 0.1 to 0.9 ($H_p/H = 0.2$) for all values Gr in case of insulated top and bottom walls, and for values of $Gr > 2.5\times10^4$ for isothermal top and bottom walls, the value of x_p/H for maximum Nu being higher for higher values of Gr. In case of isothermal top and bottom walls, for values of $Gr \leq 2.5 \times 10^4$, Nu either monotonically decreases (for moderate Grashof numbers) by a small amount, or may even pass through a minimum (for low Grashof numbers) as x_p/H is increased.
- In general, the circulation inside the enclosure initially increases as H_p/H increases ($x_p/H = 0.5$) and then tends to approach a constant value.
- As H_p/H increases ($x_p/H = 0.5$), Nu increases and approaches a constant value in case of adiabatic top and bottom horizontal walls, while Nu increases continuously in case of isothermal top and bottom walls.
- When L/H (inverse of the aspect ratio A) is increased (by keeping H constant) from 0.2 to 3.0, Nu tends to approach a constant value, having initially increased by a small amount in case of isothermal top and bottom walls and having initially decreased by a moderate amount in case of adiabatic top and bottom walls.

Turner and Flack (1980) have carried out an experimental investigation of the same problem with adiabatic top and bottom horizontal walls in the range $5\times10^6 \leq Gr \leq 9\times10^6$. In this range of enclosure height based Grashof number, Nu ($=\alpha H_p/\lambda_f$) is found to be insensitive to aspect ratio ($0.5 \leq H/L \leq 2.0$). Further, the optimum location $(x_p/H)_{opt.}$ (i.e., the location for maximum heat transfer) of the strip heater is found to be ≈ 0.6. Turner and Flack (1980) have correlated their data for the case H/L = 1.0, and $x_p/H = 0.5$ in the form:

$$Nu = A_1 \, Gr^{A2}, \tag{1}$$

where, Nu is based on the height H. At the values of Hp/H equal to 0.125, 0.25, 0.5 and 1.0, the constant A_1 takes on the values 0.045, 0.069, 0.097 and 0.119 respectively, while A_2 takes on the values 0.33, 0.32, 0.31 and 0.30. Comparison with the work of Chu and Churchill (1976) has revealed that the above correlation is applicable for values of Gr down to about 10^4. However, in electronic cooling applications, the temperature reached by the electronic chip is of primary interest because the power dissipation is known. For this reason, Flack and Turner (1980) have also correlated the data in the form:

$$Nu = B_1\, Gr^{B2}, \tag{2}$$

where, Gr is now the heat flux based Grashof number with enclosure height as the characteristic dimension. At the values of H_p/H equal to 0.125, 0.25, 0.5 and 1.0, the constant B_1 takes on the values 0.462, 0.377, 0.287 and 0.196 respectively, while B_2 takes on the values 0.25, 0.24, 0.24 and 0.23.

Data for the variation of Nu with x_p/H, applicable for the range $0.125 \leq Hp/H \leq 1.0$, has been presented graphically by Flack and Turner (1980). However, for a heater with $H_p/H = 0.133$, Chadwick et al. (1991) have suggested a relation of the type given by Eq. 2 with the values 0.883, 0.99 and 0.497 for B_1 and 0.2, 0.19 and 0.22 for B_2 at values of x_p/ H, 0.8, 0.5 and 0.2 respectively. They have also suggested that the local Nusselt numbers can be calculated using the relation:

$$Nu_y = C1Gry^{*C2}, \tag{3}$$

Where, Gry is the heat flux based Grashof number and y is the distance measured from the leading edge of the heater. At the locations x_p/H=0.2, 0.5 and 0.8, the values of C_1 are 0.438, 0.454 and 0.526, with C_2 taking 0.193, 0.199 and 0.192, respectively. It may be noted that for an isoflux vertical flat plate, C_1=0.495 and C_2=0.20.

The transient numerical simulations performed by Kuhn and Oosthuizen (1987) indicate that during the time between the power supply and the final steady state, the electronic chip may attain temperatures higher than steady state values.

A numerical and experimental study of an enclosure with multiple flush mounted heaters on a vertical wall has been performed by Chadwick et al. (1991). They have considered an air filled vertical cavity with aspect ratio A = H/L = 5 in which equally spaced multiple heaters (H_p/H = 0.133), are mounted on one vertical wall, which has adiabatic sections of Hi/H = 0.133, at the top and bottom. The opposing wall is maintained isothermal with the other walls insulated. The calculations show that the heaters forming the multiple heat sources exhibit lower average heat transfer coefficients compared to free convection from an isolated vertical flat plate with height equal to that of a heater (i.e., H_p), the isoflux flat plate correlation being $Nu = 0.594\, Gr^{0.2}$. Chadwick et al. (1991) have correlated the average heat transfer data for the range $10^4 \leq Gr \leq 5 \times 10^6$ in the form:

$$Nu = C\, Gr^{0.215}, \tag{4}$$

where, the Grashof number is heat flux based, the characteristic dimension being the heater height. The values of the coefficient C for three, four and five heater configurations (heaters counted from the bottom) are shown in Table 1.

Table 1: Values of C in the Average Heat Transfer Correlation Nu = C Gr $^{0.215}$ for Multiple Heat Sources (Hp/H = 0.133) on a Vertical Wall of an Enclosure (Chadwick et al., 1991)

Three Heat Sources $H_i/H = 0.1667$			Four Heat Sources $H_i/H = 0.0667$			Five Heat Sources $H_i/H = 0.01667$		
Heater	x_p/H	C	Heater	x_p/H	C	Heater	x_p/H	C
3	0.2	0.272	4	0.2	0.228	5	0.20	0.204
2	0.5	0.339	3	0.4	0.267	4	0.35	0.237
1	0.8	0.453	2	0.6	0.330	3	0.50	0.263
			1	0.8	0.427	2	0.65	0.318
						1	0.80	0.425

Note: The correlations are valid in the range $10^4 \leq Gr \leq 5 \times 10^6$

A numerical and experimental study similar to that of Chadwick et al. (1991) is reported by Ho and Chang (1994), wherein it is concluded that contrary to the influence of increasing Rayleigh number, the increase in the aspect ratio of the enclosure results in substantial reduction in the heat dissipation from the discrete sources. The maximum surface temperature among the four discrete heaters is found to correlate as:

$$T^*_{max} = 0.962Ra^{-0.186} A^{0.297}, \tag{5}$$

where A = H/L is the aspect ratio of the enclosure.

Papanicolaou and Gopalakrishna (1995) have studied numerically the natural convection from isoflux heat sources attached to the bottom wall of a shallow enclosure with the top wall maintained isothermal. The configuration resembles a laptop or box type computer. The results show that with discrete heat sources, there is no well-defined critical Rayleigh number for transition from conduction to convection r´egime and that the transition occurs over a range of Rayleigh number. In the presence of neighbouring heaters, the heat transfer from a heater is significantly reduced (by about 30 per cent). The results can be used to determine the heat source temperatures for a given module heat dissipation and the air gap thickness.

Experimental studies on natural convection cooling of a discrete heat source on a horizontal conduction board in a shallow enclosure have been conducted by Ortega and Lall (1998) for various positions of the board and orientations of the heat source (facing down and up). It is found that heat transfer correlations for heated horizontal surfaces are not applicable to discrete heat sources attached to cards and placed in an enclosure, because both enclosure and source length scales become important.

Dehghan and Behnia (1996) have numerically studied the effect of radiation on conjugate natural convection in a top open cavity driven by a thin iso-flux discrete heater attached to the inside surface of one of the finite thickness, externally insulated, vertical walls. Their study reveals that radiation has a significant influence on thermal and flow fields, although, in this particular problem, the enhancement of heat transfer by radiation is compensated by a weaker convective heat transfer, leaving the heater temperature unaltered from the pure convection case.
Santra et al. (1996) have performed a computational study of natural convection heat transfer from a discrete isothermal heater located on one vertical wall of a partially-open two-dimensional enclosure. The vertical wall partly faces a barrier protruding up from the floor and an opening. They have studied the effect of different heater locations and barrier heights and have concluded that the uppermost and bottommost positions are not very favourable for the placement of the heater and that the barrier heights up to 2.5 times the size of the heater have only a negligible effect on the heat transfer. Correlations are presented for calculating the heat transfer from the discrete isothermal heat source.

2.2 Liquid cooling

In the experimental study performed by Keyhani et al. (1988a), eleven isoflux heaters ($H_p/H = 45.45\times10^{-3}$) and eleven insulated spacings each of height H_i ($H_i/H = 45.45\times10^{-3}$) are alternately arranged on one vertical wall (starting from an insulated portion at the bottom of the wall) of a vertical cavity of nominal aspect ratio A = H/L = 16.5 and width aspect ratio B = W/L = 10 (W=25.4 mm), with the opposing cold vertical wall maintained isothermal. The power dissipation of each heater is varied between 5 and 50 W resulting in an enclosure height and heat flux based Rayleigh number range $2.87\times10^{9} \leq Ra \leq 2.36\times10^{10}$ with ethylene glycol ($94.3 \leq Pr \leq 161.7$, the average value of Pr being ≤ 150). Similar to differentially heated tall enclosures, primary (i.e., unicell), secondary (i.e., multicellular) and tertiary flows have been observed in the cavity. It is suggested that turbulent flow can be expected for Ray (heat flux based local Rayleigh number with y measured from the bottom of the enclosure to the mid height of any heater) values in the range $9.3 \times 10^{11} \leq Ray \leq 1.9\times10^{12}$. The local Nusselt number at the mid height of any heater can be calculated from the relation:

$$Nu_y = 1.009 Ra_y^{*0.1805}, \qquad (6)$$

Relations have also been presented for the variation of the local Nusselt number Nu with Ra, based on the air gap breadth L in the enclosure and the heater heat flux $q^{\cdot}$, for each heater, in the following form:

$$Nu = D_1 Ra^{D2}, \qquad (7)$$

with D_1 taking on the values 0.345, 0.272, 0.266, 0.218, 0.176, 0.189, 0.170, 0.168, 0.157 and 0.117 and D_2, the values, 0.242, 0.243, 0.236, 0.244, 0.255, 0.242, 0.249, 0.243, 0.243 and 0.259 respectively for the ten heaters counted from the bottom (Data has not been reported for the eleventh heater due to a problem in the thermocouple circuit for this heater). It may be seen that the largest change in Nu occurs between the first two heaters [(Nu for the second heater) ÷ (Nu for the first heater) ≈ 0.79]. Further, the average Nusselt number for discrete heating is found to be 68 to 108 per cent higher than that for complete isoflux heating of the vertical wall.

Keyhani et al. (1988b) have carried out experimental and numerical studies on a similar problem, with three heaters [$(H_s/H)=(H_i/H)=0.167$] flush mounted on a vertical wall of a cavity with aspect ratio $A = H/L = 4.5$. With ethylene glycol as the working medium, D_1 in Eq. (7) takes on the values 0.292, 0.184 and 0.195 while D_2 takes on the values 0.259, 0.253 and 0.234 for the three heaters respectively. However, for the cases of Pr = 1 and 25 ($10^3 \leq Ra \leq 4.4 \times 10^8$), values of 0.602, 0.437 and 0.239 for D_1, and 0.198, 0.202 and 0.216 for D_2, have been suggested based on computations. Comparison of the results with Keyhani et al. (1988a) indicate that a decrease in the aspect ratio of the enclosure results in a reduction in Nu, contrary to the case of a vertical cavity with complete heating on the vertical wall.

Heindel et al. (1995a) have performed two- and three- dimensional numerical analyses of natural convection arising from discrete isoflux heat sources placed on one vertical wall of an enclosure with the opposing wall maintained isothermal and the rest of the walls insulated. The two-dimensional study considers 3 discrete heaters along the height of one vertical wall while the three-dimensional analysis considers a 3 × 3 array of heaters for $10^5 \leq Ra \leq 10^8$, $0.2 \leq W^*_p \leq 4.0$, Prandtl number Pr = 5 and $H^*_e = 7.5$. The quantities W^*_p and H^*_e are the dimensionless heater width and enclosure height, the characteristic dimension being the heater height. The Rayleigh number is based on heater height and heat flux. The computations reveal that three-dimensional effects become significant only for $W^*_p \leq 3$ and that two-dimensional results, although may slightly over predict the heater temperatures, are satisfactory as long as $W^*_p > 3$. For $W^*_p = 1$, while the three-dimensional computations showing one-fourth power dependence of Nusselt number on Rayleigh number are in excellent agreement with the experimental results (Heindel et al., 1995b), the two-dimensional analysis under predicts the Nusselt number (i.e., overpredicts the heater temperature) by about 20 per cent.

3 Discrete heat sources embedded in a substrate

Joshi et al. (1993) have reported the results of a conjugate natural convection numerical study of a heat-generating package embedded inside a substrate such that the heater surface is flush with that of the substrate. The substrate and protrusion assembly is placed in vertical position in a liquid-filled enclosure with isothermal walls. The study shows that the heat source temperature does not undergo significant

changes when Prandtl number is between 7 and 100, $\lambda^*_p > 25$ and $\lambda^*_s > 10$, where λ^*_p and λ^*_s are the package-to- and the substrate-to-fluid thermal conductivity ratios.
Heindel et al. (1995c) have numerically studied the case in which one vertical wall of the enclosure is composed of a 3 array of discrete high thermal conductivity blocks embedded in a substrate across the full thickness in a regular pattern. Isoflux heating is applied to the back faces of the blocks while the front face consisting partly the substrate and partly the blocks are in contact with a liquid. The wall facing the substrate is maintained isothermal.
The Prandtl number Pr is taken as 5 and 25 corresponding to water and fluorinert liquid FC-77. The block-to-fluid thermal conductivity ratio λ^*_p and the substrate-to-fluid thermal conductivity ratio λ^*_s are taken as 650 and 0.48 corresponding to water, G-10 fibreglass (λ = 0.29 W.m^{-1}.K^{-1}) and copper (λ = 400 W.m^{-1}.K^{-1}), while with FC-77, λ^*_p and λ^*_s are 6420 and 6.7. The Rayleigh number Ra (based on heater height and heat flux) is varied between 10^5 and 10^8. The fluid flow adjacent to the blocks reveals noticeable three dimensional effects and the discrete heating affects the local heat transfer substantially with large spikes in the local heat flux near the block edges. Higher values of λ^*_s, while leading to lower block temperatures also make the flow and temperature fields increasingly two dimensional as result of which the fluid circulation tends to encompass the entire width of the enclosure. Detailed two-dimensional numerical results obtained by Heindel et al. (1995d) have shown that with increasing Rayleigh number, the core region begins exhibit higher thermal stratification and multiple fluid cells, while the substrate conduction decreases. At λ^*_s = 10, the discrete heaters begin to lose their thermal identity and at $\lambda^*_s \approx 10^3$, the flow and temperature fields resemble those of a differentially heated cavity. At lower values of Ra, the intensity of fluid motion decreases with increasing λ^*_s, while the opposite happens at higher Rayleigh number. An increase in λ^*_s is found to promote thermal spreading thereby reducing the differences between the average surface temperatures of the three heaters as well as the maximum heater surface temperature.

3.1 Heat-sink coupled substrate-embedded discrete sources

For the enclosure with 3×3 array of blocks embedded in the substrate, Heindel et al. (1996a) have found that the use of heat sinks attached to the blocks as well as the opposing cooled wall leads to a heat transfer enhancement by a factor of 24 for the vertical and 15 for the horizontal (bottom-heated) configurations with corresponding thermal resistances of approximately 2 cm^2.^{0}C.W^{-1}. The horizontal configuration results in greater uniformity of the block temperatures. Treating the heat sink as a porous medium with Brinkman-Forchheimer-extended Darcy model is found to yield satisfactory numerical predictions, compared to Brinkman-extended Darcy model.
Yu and Joshi (2002) have studied the heat transfer enhancement of a pin-fin array attached to a thin foil heater embedded in a substrate forming one wall of an enclosure. The back side of the substrate is insulated while the external surfaces of the other walls are maintained isothermal. Results without the heat sink have shown that radiation accounts for 40 per cent of the overall heat transfer and even more at

lower power levels. Horizontal arrangement with open top is found to be better by a factor of 2 than the vertical arrangement with open side. However, in case of complete enclosure, the vertical arrangement produces lower thermal resistance. The heat transfer performance of the enclosed heater with heat sink, although better than that of bare heater exposed to ambient, is not much pronounced, due to the shrouding effect of the enclosure walls. The reduction in thermal resistance with pin-fin arrays is more significant for top-open horizontal enclosures, reaching about 20 percent with 40 per cent opening.

4 Protruding discrete heat sources in enclosures

In the numerical study performed by Ju and Chen (1996), five equally spaced heat generating protruding heaters ($H_p/H = 1/11$) are mounted to an insulated wall. The protrusion of the heaters is 1/5 of the width of the air-filled enclosure (i.e., $L_3/L=1/5$). Equal spacings are left at the top and bottom of the vertical wall and in between the heaters ($H_i/H=1/11$). Their studies indicate that the local Nusselt number Nu_y correlates with the heat generation based Rayleigh number Ra_y as:

$$Nu_y = 0.137 Ra_y^{0.224}, \tag{8}$$

for an aspect ratio H/L=4.58, y having been measured from the bottom. This result derived from computations and aimed at verifying the measurements of Keyhani et al. (1991) is in almost exact agreement with the correlation derived from experimental data.

Sathe and Joshi (1991, 1992) have performed a conjugate two-dimensional numerical study of the natural convection liquid immersion cooling of a heat-generating protrusion of height H_p and thickness L_p attached to a substrate of height H_s and thickness L_s. The substrate and protrusion assembly is placed in vertical position in a liquid-filled enclosure of dimensions H_e and L_e with isothermal walls. The heights corresponding to floor to substrate bottom, substrate bottom to protrusion bottom, protrusion top to substrate top are respectively H_1, H_2, H_3 and H_4 respectively with the vertical face of the substrate in contact with the protrusion coinciding with the vertical mid-plane of the enclosure. The protrusion height Hp is taken as the characteristic dimension. The characteristic temperature difference is based on the heat generation rate per unit width (or wattage/unit width) and the fluid thermal conductivity (i.e., $\Delta T = \dot{Q}_W/\lambda_f$). The numerical results show that the solid conduction effects must be included in the model, that simplistic solid surface thermal conditions are inappropriate, that liquid cooling is preferable to air cooling only when protrusion internal thermal resistance is small (i.e., $\lambda^*_p > 1$), that the substrate height has a marginal effect on the maximum temperature of the protrusion and that a slight cooling enhancement of the order of 10 per cent occurs when L^*_p increases from 0.25 to 1.25. The maximum dimensionless temperature of the package is correlated as:

$$T^*_{max} = [(0.1\,\lambda^*_p{}^{-0.86})^{3/2} + (0.277Ra^{-0.133})^{3/2}]^{2/3}$$
$$[0.6 + 0.9\exp(-0.1\,\lambda_s{}^*)]\,[1.2 - 0.3L_p{}^* + 0.1L_p{}^{*\,2}] \quad (9)$$

in the range $10^5 < Ra < 10^7$, $0.1 < \lambda^*_s < 10^3$, $0.1 < \lambda_p{}^* < 10^3$, $0.25 < 1L^*_p < 1.25$, $0.3 < L\_s < 3.0$, $H_1{}^* \geq 4.5$, $H_2{}^* \geq 2$, $H_3{}^* \geq 1$, $H_4{}^* \geq 4.5$ and $Pr \geq 10$, where $\lambda_p{}^*$ and $\lambda_s{}^*$ are the protrusion-to- and the substrate-to- fluid thermal conductivity ratios. The protrusion temperature does not change significantly when any one of the walls is changed to adiabatic condition. With the top boundary isothermal and other boundaries adiabatic, the protrusion temperature increases by as much as 28 per cent. Wroblewski and Joshi (1993) have numerically studied the liquid natural convection cooling of a rectangular electronic package of height H_p, thickness L_p and width W_p attached to a substrate of height H_e and width W_e with the substrate forming one vertical wall of the enclosure of breadth Le, taking into account the conjugate heat transfer in the package and the substrate. The bounding surface facing the substrate is taken as isothermal and the remaining external surfaces are taken as adiabatic. The package-to- and the substrate-to fluid thermal conductivity ratios ($\lambda_p{}^*$ and $\lambda_s{}^*$) are taken as 575 and 2360 corresponding to a silicon chip, alumina ceramic substrate and fluorinert liquid FC-75 (Prandtl number=25). The package height Hp is taken as the characteristic dimension. The characteristic temperature difference is based on the heat generation rate (or wattage) and the fluid thermal conductivity (i.e., $\Delta T = Q^{\cdot}/(Hp\,\lambda_s)$). The Rayleigh number Ra is varied between 10^3 and 10^9 (A 1 cm square package with 1.5 W results in a Rayleigh number of 10^9). 75-90 per cent of the heat generated in the package is found to be transferred to the substrate from where the heat is transferred by natural convection to the opposing wall. Three-dimensional effects are found to be predominant at high Rayleigh numbers when a strong plume is generated above the package. Comparison of experimentally measured temperatures of Joshi and Paje (1991) with the numerical values reveals an agreement between -9 and 15 per cent. In the experiments use is made of 1.9 mm thick, 8.9 mm square, 20-pin leadless chip carrier package fixed to the substrate through 20 solder joints and containing a 0.4 mm thick, 1.52 mm square silicon chip. Numerically determined chip temperatures are 5 and 16 per cent lower with the top wall cooled and with both top and opposing walls cooled, compared to the case of substrate-opposing wall alone cooled (Wroblewski and Joshi, 1994). The maximum dimensionless temperature of the package, with $S_p{}^*$ denoting the dimensionless surface area of the package, is correlated as:

$$T^*_{max} = 102.5\exp[0.0291\,\lambda_s{}^{*0.45}\,(We^{*\,0.45} - 5.1^{0.45})]$$
$$[(0.0455\,\lambda_p{}^{*\,-1.038}\,p\,Ra^{-0.003})^{3/4} + (0.518Ra^{-0.25} + 0.0044)^{3/4}]^{4/3}$$
$$[(0.0062 + 0.242\,S_p{}^{*-0.546}\,\lambda_s{}^{*-0.589})^{-2.2}(0.0433\,S_p{}^{*-0.674})^{-2.2}]^{-1/2.2}, \quad (10)$$

in the range $10^3 < Ra < 10^9$, $0.5 < \lambda_s{}^* < 575$, $0.1 < \lambda_p{}^* < 2360$, $1 < W_p{}^* < 4.5$, $2.55 < W_e{}^* < 7.65$, $0.05 < L_p < 0.4$, $10 < P_r < 1000$, for $H_e{}^* = L_e{}^* = 5.1$ and $W_e{}^* > W_p{}^*$.

In a two-dimensional computational investigation devoted to conjugate buoyancy-driven flow in a liquid-filled enclosure, Heindel et al. (1996b) have considered protruding discrete packages attached to a vertical substrate forming one vertical wall of the enclosure. The heat flux from the chip is modelled as volumetric heat generation in a thin layer at the back of the package. Contact resistance between the chip and substrate is modelled as a thin layer of extra material with variable thermal conductivity. Heat fluxes in the range 3.5×10^{-6} W.cm^{-2} to 0.35 W.cm^{-2} corresponding to a Rayleigh number range of $10^4 \leq Ra \leq 10^9$, contact resistances of 0.003-10 cm^2. ^{0}C.W^{-1}, thermal conductivities encompassing materials from glass to silicon (1.48-148 W.m^{-1}.K^{-1}) and fluorinert dielectric liquid FC-77 (Prandtl number =25) are considered with the package-to-fluid thermal conductivity ratio λ_p^* fixed at 2350 and the substrate-to-fluid thermal conductivity ratio λ_s^* varying between 23.5 and 2350. The Rayleigh number is based on package height and heat flux. As the Rayleigh number increases, more fluid penetrates the regions between the protrusions and the package-to-fluid heat transfer increases. The effect of contact resistance is minimal in the range considered. At values of $\lambda_s^* < 47$, the substrate shows guard-heater effect, increasing the percentage of applied power transmitted through the package into the liquid.

In the three-dimensional computational study performed by Liu et al. (1997), a vertical, horizontal or a staggered array of protruding heat generating elements is attached to substrate which forms an externally insulated finite thickness vertical wall of an enclosure. The wall facing the substrate is an infinitesimally thin isothermal cold wall. The rest of the four walls are also infinitesimally thin, but are insulated. The maximum temperature for the vertical array correlates as $T^*_{max} = 7.3074Ra^{-0.1478}/\ \lambda_p^{*0.02377}\ [0.00009421\ (\ln \lambda_s^*)^2 - 0.012563 \ln \lambda_s^* + 0.062863]$ in the range $1.7 \times 10^5 < Ra < 8.6 \times 10^5$ (volumetric heat generation rate and heater height based Rayleigh number), $0.209 < \lambda_s^* < 20.9$ (substrate-to-fluid thermal conductivity ratio) and $1.57 < \lambda_p^* < 157$ (protrusion-to-fluid thermal conductivity ratio).
A three-dimensional computational study is performed by Baek et al. (1997), considering an enclosure similar to that of Liu et al. (1997) except that a single protruding heat source is attached to the substrate. By taking into account the effect of radiation on conjugate natural convection, the authors have shown that the substrate thermal conduction, radiation and the emissivity of the walls, all have significant effect upon the temperature rise of the heater.
Liu et al. (1988) have numerically investigated the natural convection in a rectangular parallelepiped enclosure of dimensions L × W × H (L = 9 mm, 18 mm or 30 mm, W=144 mm, H=120 mm) with the top and bottom walls (of dimensions L × W) isothermal (at 21.1 ^{0}C), and three of the vertical walls, two of dimensions H × L and one of dimensions H × W, insulated. On the remaining insulated vertical wall, nine heaters (a 3 × 3 array), each of power dissipation 0.4 W, height H_p=24 mm, width W_p=8 mm and thickness L_p=6 mm are mounted in three rows with uniform spacings in between. Calculations have been performed taking into account the variable thermophysical properties of Fluorinert FC 75 liquid. The Rayleigh number

based on heater height and heat flux is Ra = 8.7×10^6. The flow and temperature fields are found to oscillate periodically. The top row heaters are found to reach the maximum temperature (42 ^{0}C for L = 30 mm, 46 ^{0}C for L = 18 mm and 64 ^{0}C for L = 9 mm). The bottom row heaters have the lowest temperature of about 30 ^{0}C.

5 Conclusions

Earlier, most of the analytical and numerical studies have neglected the influence of heat conduction in walls and other elements on natural convection in enclosures in order to render the problems tractable. It is now generally recognized that for obtaining accurate results, the coupling of conduction with natural convection should be taken into account and that the use of simplistic boundary conditions in the interest simplifying the problems is inappropriate.

Since the temperature differences encountered in natural convection applications are usually not large, the effect of radiation on natural convection was mostly neglected prior to 1990's. However, some studies have pointed out that the effect of radiation, even on natural convection, is likely to be significant in geometries where non-uniform and unsymmetrical temperature distributions occur.

Almost no attempts have been made to compare the results of two and three dimensional numerical analyses of natural convection, taking radiation into account. Such studies are important to assess the amount of deviation that is likely to occur if two-dimensional approximations are made for computational economy.

The work reported on natural convection in discretely heated enclosures is mostly confined to enclosures of high aspect ratios. Much work remains to be done in respect of medium and low aspect ratio enclosures and with varying position of the heaters.

Turbulent natural convection studies for enclosures with discrete heat sources offer a great deal of scope for further research.

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