Experimental Approach of Natural Convection Heat Transfer in Vertical Channel with Horizontal Heated Plate at Small Height Ratio

Anu Nair.P[1], Dr.K.Karuppasamy[2]

[1]PG Scholar
Department of Mechanical Engineering
Regional Center of AnnaUniversity,Tirunelveli,Tamilnadu
anunair67@gmail.com

[2]Assistant Professor
Department of MechanicalEngineering
RegionalCenterofAnnaUniversity, Tirunelveli,Tamilnadu
kkssurabu2001@yahoo.co.in

ABSTRACT---*The thermal problems of most electronic components have been solved in recent year. Passive cooling of electronic component by Free convection heat transfer has major advantage of no vibration, least expensive and most authentic method of heat rejection. In many numerical and experimental works, heat source location is studied, especially with two or more sources on a vertical wall. The parameters frequently considered are: the Rayleigh number, the distance between heat sources. In this work two experiments were conducted. viz. natural convection in a horizontal plate and natural convection in a vertical channel with single heat source. The experimental setup has been designed and fabricated. In the first case, Experimentally and Numerically the convective heat transfer coefficient were calculated and compared with the value available from correlation and found to be in good agreement. In the second case, Experimentally and Numerically, heat transfer coefficient were calculated at small height ratio and to determine the optimum location of heat source in vertical channel.*

Keywords--- Natural Convection, Horizontal Plate, Height Ratio.

1. INTRODUCTION

Natural convection flow with internal object in vertical channel is encountered in several technological application such as heat dissipation in electronic circuits, refrigerators, heat exchangers, nuclear reactor fuel elements, and dry cooling towers, where power distribution and location of discrete heat source are very important, natural convection is the only mode of heat transfer mode in the case of artificial cooling failure. In these equipment's, the source of heating, in general is either due to volumetric heat generation or due to surface heat fluxes. For instance, electronic equipment generates heat, which can be expressed in terms of volumetric heat generation. Volumetric heat generation in nuclear fuel rods is due to nuclear reaction. The present work is related to natural convection in a typical geometry of an electronic chip undergoing heating. From heat transfer point of view, the electronic chip is modeled as a heat source. In this geometry, the location of heat source placed an important role in the natural convection heat transfer. Detail knowledge of heat transfer characteristics of such geometries is an essential requirement for the optimal design of these equipments. The objective of the present work is to experimentally obtain an optimal location for a single heat source.

A comprehensive review of natural convection in electronic cooling is available in [1]. In literature, although plenty of studies on partitioned enclosures and discrete heat sources attached to adiabatic walls are encountered, natural convection from heated elements within enclosures has received very little attention until recently.

Electronic devices are progressively miniaturized, but the performances are continuously improved. This trend has led to the production of chips with greater power density, and caused some undesired problems, such as shortened life time, malfunction and the failure of the devices. Efficient heat dissipation is therefore very important for electronic devices. Without suitable thermal management, the overheating would significantly affect the component reliability through, for example, thermal shock, electro migration or material failure. For instance, the reliability of a silicon chip declines about 10% for every 2 ^{o}C rise in temperature [1]. Yeh [2] pointed out that over 50% of electronic failures are temperature-related in a US Air Force study. Accordingly, lots of researches have been conducted into improving the thermal design of electronic devices.

The optimization of the heat transfer has increased importance in electronic packaging due to the higher heat fluxes and to the electronic components and equipments miniaturization [3]. In spite of the abundant results about natural convection in electronic packaging, work dealing with heat transfer maximization is scarce in literature [4,5]. The opportunity to optimize the positions of concentrated heat sources was recognized in [6, 7]. The parameters frequently considered are-the Rayleigh number, the distance between heat sources and the ratio between heat source dissipation rates. [8,9,10]. In many numerical and experimental works, the effect of location of heat source is studied, especially with two or more sources [11,12,13].Previous investigator has analyzed a similar configuration as that of the present case by considering cooling of right wall alone. [14].

Da Silva et al. [15] studied the optimal heat source distribution on an adiabatic wall. They carried out a theoretical analysis and numerical simulations of the natural convection in a vertical cavity. In the work, curves showing the heat source optimal location as function of Rayleigh number are presented for cavities with one, two and three heat sources. To obtain the optimal locations they extensive simulated all combinations between the control parameters, Rayleigh number and heat source locations. They concluded, contrary to the results presented by Liu and Phan-Thien [16], that the optimal arrangement is not described by a constant ratio between the centerto- center distance between heat sources, but by a function that depends strongly on the Rayleigh number and the heat source dimension.

Dagtekin and Oztop [17] studied the natural convection heat transfer and fluid flow of two heated partitions attached to the lower adiabatic wall for Ra numbers of 10^4–10^6.Enclosure is cooled from the left and top walls while the only heat source within the medium is due to the partitions whose length and positions were varied. They showed that the mean Nusselt number increases with the partition length. Merrikh and Lage [18] studied the case of an enclosure heated from the side and containing equally spaced, conducting solid square blocks by using a continuum model, which treats the fluid and solid constituents individually. The dispersive effect of the solid constituent is isolated by increasing the number of solid blocks while reducing their size as to maintain their relative total volume constant. Results obtained for a wide Ra number range of 10^5–10^8 and several values of solid-to-fluid conductivity ratio. Desai et al. [19] considered rectangular enclosures with multiple protruding heaters mounted on one side wall, with the top wall being cooled, and the opposing vertical wall and the bottom wall being insulated.

The purpose of this paper is to find optimum location of single heat source in vertical channel. In this work we present a study of two –dimensional, natural convection flow and steady-state heat transfer in vertical channel with horizontal heated plate is determined experimentally and analysed numerically using FLUENT software. [21].

2. EXPERIMENTAL METHODOLOGY

2.1. Investigation Of Free Convection Heat Transfer In A Horizontal Plate

As a first step, a validation experiments has been conducted by studying the heat transfer in an isolated heated horizontal plate. The details of the experiment setup is shown in Table 1

Table 1:Details of Experimental Set-up

SI NO	Description	Dimension/Range
1	Rectangular Horizontal SS Plate (1 no.)	250×50× 3mm
2	Rectangular Box with Thermocol (2 no.)	500 x 500x 150 mm
3	Heat Input	0-300 W
4	Single Phase Closed Type Dimmerstat	0-1 Amps
5	Ammeter	0-1 Amps
6	Multimeter	350 V_{AC}
7	K-type Thermocouple with indicator	0-1000°C

Experimental apparatus has been specially planned and formulated to carry out investigations on electronics devices. The experimental setup consist of an apparatus, temperature indicator, K –type thermocouple, Multimeter, Ammeter, and AC power supply whose schematic diagram is as shown in Figure 1.

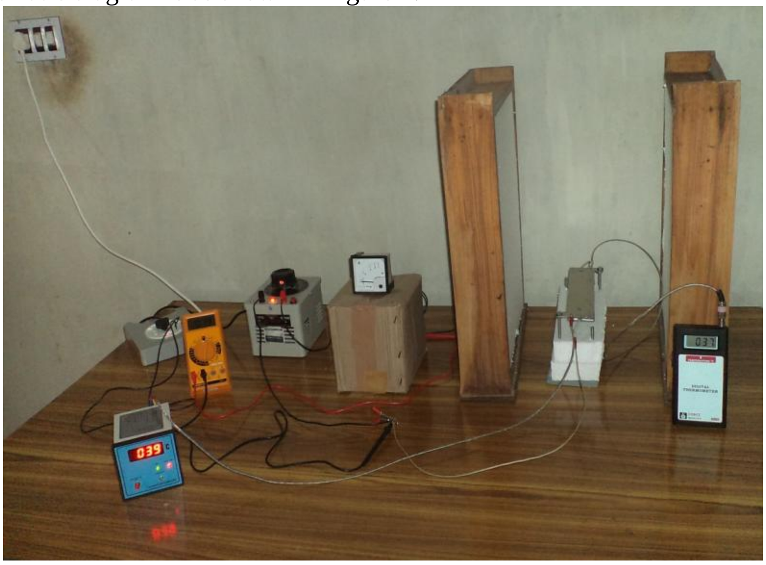


Figure 1: Photographic View of the Experimental setup

The apparatus consists of a stainless steel plate and flat heater fitted on a support structure in a horizontal fashion. The heat input to the heater is measured by an ammeter and a multimeter and is varied by a dimmerstat.Ammeter are connected in series manner and multimeter are connected in Parallel manner. The two rectangular box are kept as adiabatic walls which is filled with thermocol to avoid the external disturbance on both sides and only heat is supplied to the vertical plate Temperatures is measured using K-type thermocouple. Thermocouples is fixed in between the flat heater and stainless steel plate with the help of blind holes by using the nuts and bolt on the support structure and taken out of the plate. The K type thermocouples are connected to the temperature indicator to measure the temperature of the plate. The stainless steel plate has been polished to minimize the radiation losses. As it could be seen there are 2 holes on the plate which house the screws that are used in fastening the plate and the heater together. The parameter varied during the experimentation is heat input to the heater. The whole assembly was highly polished on its outer surface to obtain an emissivity of 0.17

2.1.1 *Procedure for the Validation Experiment*

The Horizontal heat source with the thermocouples is placed centrally inside two rectangular boxes, to isolate the experiment from external disturbances. A stabilized power input through a regulated AC power supply is supplied to the heater. All measurements are done after the system reached steady state conditions, which typically took one hour for each set of reading. The mean of the temperature of the central plate so obtained is made use of for the computation of the radiative and convective heat transfer rates, and the average heat transfer coefficient. The experiment is repeated for different power inputs, i.e., for the different temperatures of the central plate.

The analysis was based on the method of calorimetry i.e.

$$Q_C = Power - Q_R$$

$$Q_R = \varepsilon \sigma A\left(T_h^4 - T_\infty^4\right)$$

$$h = \frac{Q_C}{A \Delta T}$$

2.1.2 *Numerical Analysis for the Free Convection Heat Transfer In A Horizontal Plate*

Figure 2: Boundary Condition

The CFD package, FLUENT 14.0 was used in the two dimensional simulation of natural convection of the experimental setup. The geometry of the problem is shown in Figure 2. It is modeled in the same dimension as that of the experimental Set-up. The fluid medium (air)is assumed to be Boussinesq approximation with constant thermo physical Properties. Temperature variations are small when compared with the absolute temperature, and, consequently, we assume that the Boussinesq approximation is valid.The zero heat flux boundary condition is applied at the right and left side of the plate , Heat flux boundary condition is applied at the bottom portion,andAmbient air is exerted at the top. The finite volume technique and the semi-implicit method for the pressure-linked equation (SIMPLE) are used to solve the basic conservation equations.The conservation equations for continuity, momentum and energy are

$$\frac{\partial U}{\partial X}+\frac{\partial V}{\partial Y}=0$$

$$\rho\left(U\frac{\partial U}{\partial X}+V\frac{\partial V}{\partial Y}\right)=-\frac{\partial P}{\partial x}+\mu\nabla^2 U$$

$$\rho\left(U\frac{\partial V}{\partial X}+V\frac{\partial V}{\partial Y}\right)=\frac{\partial P}{\partial Y}+\mu\nabla^2 V+\rho g\beta\left((T-T_0\right)$$

$$\rho c_p\left(u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}\right)=k\nabla^2 T$$

$$\nabla^2=\frac{\partial^2}{\partial x^2}+\frac{\partial^2}{\partial y^2}$$

2.2 Investigation of Free Convection Heat Transfer in Vertical Channel With Horizontal Heated Plate at Different Height Ratio

Figure 3: Photographic View of Experimental Setup at Different height Ratio in vertical channel

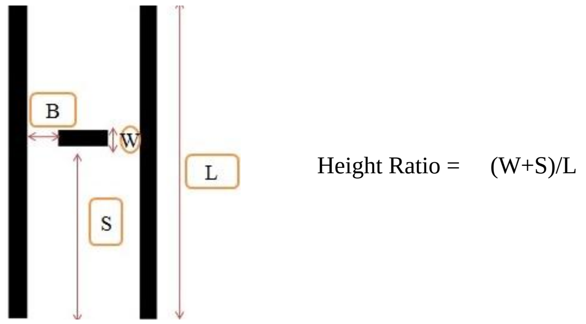


Figure 4: Line Diagram of Experimental Setup at Different height Ratio in vertical channel

2.2.1 Procedure For Free Convection Heat Transfer In Vertical Channel With Horizontal Heated Plate At Different Height Ratio

The first step is to fix the gap between the heat source and the vertical channel in the enclosure.The parameter varied during the experimentation are Height ratio and heat input. Thermocol help to increase or decrease the height of the horizontal heat source.Ac power supply is given to the heat source through the Dimmerstat. All measurements are done after the system reached steady state conditions, which typically took three hour for each set of reading.The experiment was repeated for different height ratio and power input.The photographic view and line diagram of experimental set at different height ratio in vertical channel is given in Figure 3 and Figure 4.

2.2.2 Numerical Analysis For The Free Convection Heat Transfer In A Vertical Channel With Horizontal Heated Plate

The CFD package, FLUENT 14.0 was used in the two dimensional simulation of natural convection of the experimental setup. The geometry of the problem is shown in Figure 5. It is modeled in the same dimension as that of the experimental setup. It consists of a heat source placing centrally between two adiabatic wall. This whole assembly is placed inside a large enclosure to simulate natural convection. The enclosure is filled with a Newtonian fluid. The walls of the enclosure are kept as adiabatic. The pressure outlet boundary condition is applied to bottom and top opening of the enclosure .In these inlet and outlet boundaries pressure is given as atmospheric pressure. adiabatic condition is given to two side end walls. The Adiabatic side plates on both sides of the heat source are modeled as a rectangular solid with adiabatic boundary condition to the wall. From heat transfer point of view chip is modeled as heat source. Volumetric heat generation rate Q in Wm^{-3} is given to solid heat source. Fluid is treated as incompressible with constant density.

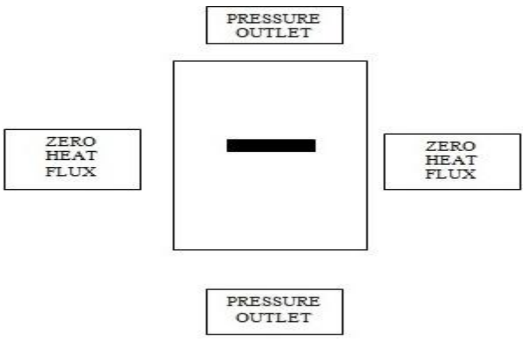


Figure 5: Boundary Conditions

2.3 Numerical Procedure

The governing Differential Equation of heat transfer and fluid flow is solved using control volume approach. Pressure based segregated solver is selected for the present computation. A first order winding is for numerical treatment of convection terms. The upwind formulation was used in order to handle the inherent instabilities associated with flow regions where convection is stronger than diffusion.. The coupling between velocity and pressure is resolved by selecting Simple algorithm. The under relaxation factors used in the present study are1 for density,1 for body force, 0.7 for momentum,0.3 for pressure and 0.9 for energy. Convergence of the solution is checked by examining the residues of discretized conservation equations of energy, mass and momentum[22]. The iteration is terminated when the maximum of all residues reaches less than $1*10^{-6}$

3. RESULTS AND DISCUSSION

3.1 Validation Experiment

The problem of free convection in a horizontal plate was studied experimentally. Experimental results for natural convection in a horizontal heat source are compared with the Numerical and standard correlation(Fishenden and Yousef) for horizontal plate facing upward in laminar region are given in Table 2

Table2: Correlation for Various Researchers

Researchers	Correlations
Fishenden and Saunders[21]	$Nu=0.540Ra^{1/4}$
Yousef et al[21]	$Nu=0.622Ra^{1/4}$

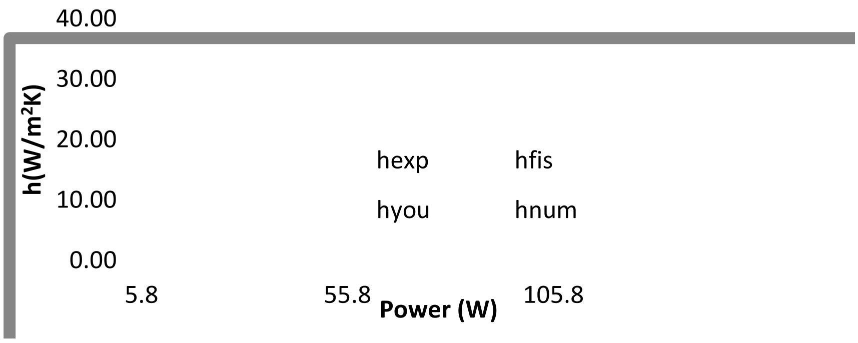


Figure 6: Heat Transfer Coefficient Vs Input Power

Figure 6 shows the variation of heat transfer coefficient with input power by experiment, numerical and correlations. When input power of heater increases convective heat transfer coefficient by experiment increases. When input power increases, it will increase the temperature of the horizontal plate, which increases the heat transfer coefficient and there by increases the Nusselt number. When power input increases, the convective heat transfer coefficient by correlation and numerical shows a downward trend. This is because of two reason (1)the horizontal dimension of the heat source is small the viscous forces try to predominate over the buoyant forces, causing a Rayleigh number decreases and therefore decrease in Nusselt number (2) the temperature increases beyond a particular value .The Velocity and Temperature contours is solved in fluent are shown in Figure 7.

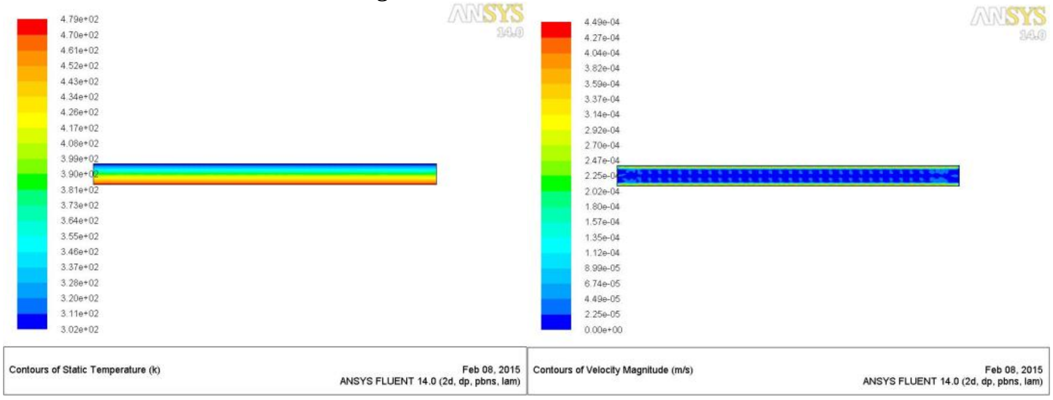


Figure 7: Temperature and velocity contour of Horizontal Plate

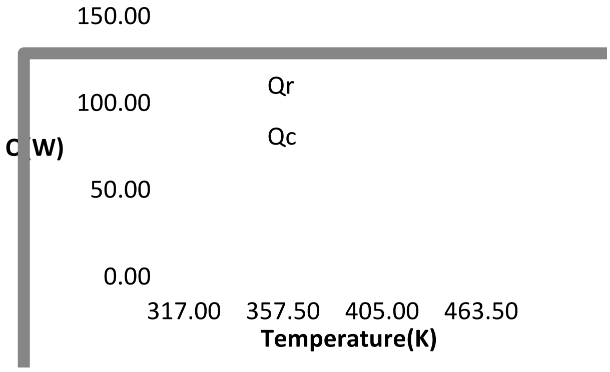


Figure 9: Heat Transfer Rate Vs Plate Temperature

Figure 9shows the radiation heat transfer and convective heat transfer is depending strongly on surface temperature of the plate. The energy emitted depends on (1)the temperature of the body and(2)nature of radiating surface of the body. BothConvective and Radiative heat transfer increases non linearly with increase in plate temperature.. At low temperature,radiation may be significant.So we expect that radiation heat transfer is directly proportional to temperature(i.e, Q α T^n for free convection, where 1.2<n<1.33 and Q α T^4 for radiation)

3.2 Investigation Of Free Convection In A Vertical Channel With Horizontal Heat Source At Different Height Ratio

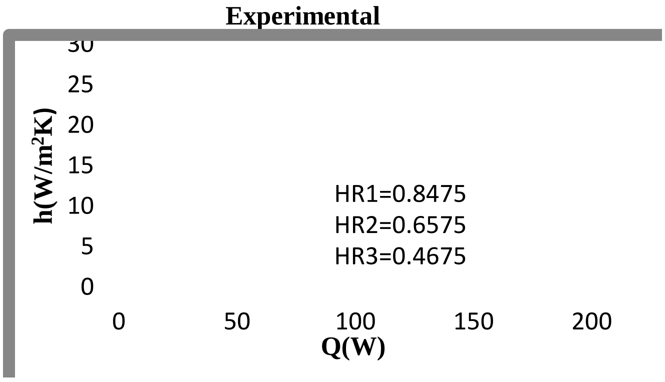


Figure 10: Heat Transfer Coefficient Vs Power by Experimental

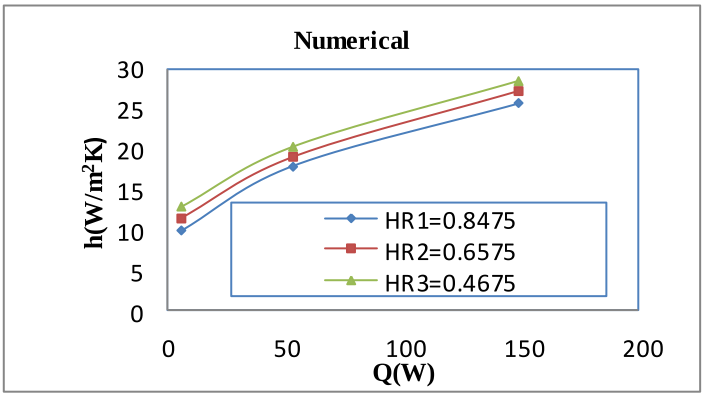


Figure 11: Heat Transfer Coefficient Vs Power by Numerical

Figure 10 and Figure 11 shows the Variation of Heat Transfer Coefficient with Power input by experimental and numerical. It has been observed that with the presence of vertical channel Heat transfer coefficient increases with increase in power input due to chimney effect. The maximum heat transfer coefficient is obtained at lower height ratio.the. Therefore the horizontal heat source should be placed only at the bottom of the channel so as to have maximum heat transfer. Streamline and Velocity contour is solved in Fluent is shown in Figure 12.

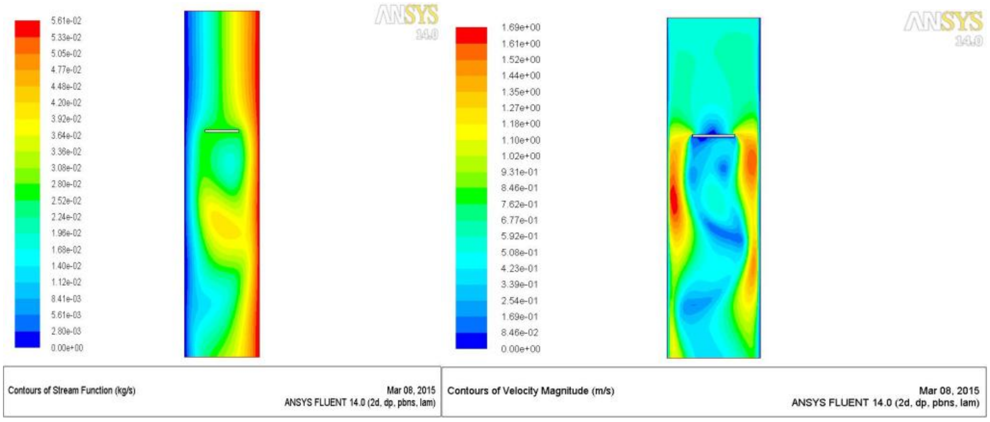
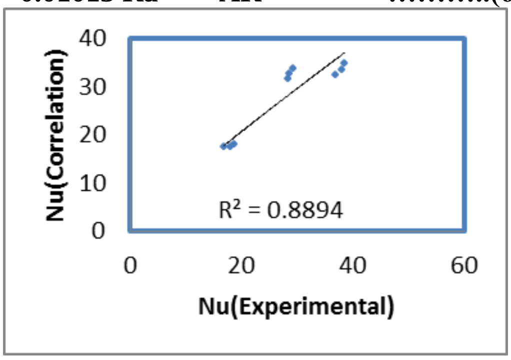


Figure 12: Stream line and Velocity Contours of horizontal plate in vertical channel**.**

A correlation for Nusselt number in terms of Rayleigh number and aspect ratio was developed using the Excel software as shown in Equation 6. The average R^2 value is found to be 0.89 shows closeness of the experimental values with the correlation.

$$\mathbf{Nu = 0.01013\ Ra^{0.60245} AR^{-0.11929}} \quad \text{...........(6)}$$



4. UNCERTAINTY ANALYSIS

All the measurement devices used in the present study are calibrated with standard instruments. The uncertainties involved in the measurement of temperature, voltage and current are given in Table 3.

Table 3: Uncertainities in the Measured Quantities

Quantity	Uncertainity(%)	Units
Temperature	± 3	^{0}C

Table 4 Uncertainities in the Derived Quantities

Quantity	Uncertainity(%)
Nusslet Number	± 5

The uncertainties in the derived quantities are obtained using the relation

$$\Delta y = \sqrt{\sum\left(\frac{\partial y}{\partial x_i}\Delta x_i\right)^2} \quad \text{...................(7)}$$

Where X_i is the measured quantity and Y the derived quantity and Δx and Δy are the uncertainties in the measured and derived quantities respectively. Based on Eqn.7, the uncertainties in the derived quantities are determined and these are reported in Tab. 4.

5. CONCLUSION

The Validation of Natural convection heat transfer in a horizontal plate and investigation of natural convection in vertical channel with heat source have been conducted. In the first case, the convective heat transfer coefficient were determined experimentally and numerically and compared with the value available from correlation and found to be in good agreement. The second experiment shows that the Maximum amount of heat transfer occur when horizontal heat source is placed at the bottom portion of the vertical channel by experimentally and numerically.

6. FUTURE ENHANCEMENT

- The same experiment can be conducted after coating the surface of heat source with paint in order to emphasize radiation effect.
- Heat source can be supported side way in order to reduce flow disturbance.
- Experiment can be conducted by tilting the entire geometry.

7. NOMENCLATURE

Q_c- Heat Transfer by Convection,W

Q_r- Heat transfer by Radiation,W

T_∞- Ambient Temperature,K

T - Plate Temperature ,K

A- Surface Area.m^2

Ra- Rayleigh Number

L- Characteristic Dimension of the Surface ,M

K- Thermal Conductivity of the Fluid,W/mK

C_p- Specific Heat at Constant Pressure,kj/kgK

ρ - Density of Fluid,kg/m3

U,V- Velocity vectors,m/s

X,Y- Cartesian Coordinates ,m

h_{Exp} - Convective Heat Transfer Coefficient by Experimental,W/m^2K

h_{Fis}- Convective Heat Transfer Coefficient by Fishenden correlation, W/m^2 K

h_{You} - Convective Heat Transfer Coefficient by Yousef Correlation, W/m^2K

h_{Num}- Convective Heat Transfer Coefficient by Numerical, W/m^2K

Greek Symbols

σ - Stefan-Botzmann Constant

ε - Emmisivity of the SS plate

γ - Kinematic Viscosity of the Fluid

β -Coefficient of Volumetric Expansion of the Fluid

α - Thermal Diffusivity of the Fluid

8. REFERENCES

[1] A. Bar-Cohen, A.D. Kraus, S.F. Davidson, "Thermal frontiers in the design and packaging of microelectronic equipment, " J. Mech. Eng. ,Vol.105 (6),pp.no 53–59,1983.
[2] L.T. Yeh, "Review of heat transfer technologies in electronic equipment, "ASME J. Electron. Packaging ,vol.117,pp.no. 333–339.1995.
[3] S. Sathe, B. Sammakia, "A review of recent developments in some practical aspects of air-cooled electronic packages, " Trans. ASME J.,vol 115,pp no 250-256,1996.
[4] Y. Liu, N. Phan-Thien, C.W. Leung, T.L. K, "An optimum spacing problem for five chips on a horizontal substrate in a vertically insulated enclosure, " Comput. Mech. 24 ,pp.no 310–317,1999.
[5] M.D. Landon, A. Campo, "Optimal shape for laminar natural convective cavities containing air and heated from the side, " Int. Commun. Heat Mass Transfer ,vol.26, pp.no 389–398,1999.
[6] I. Sezai, A.A. Mohamad, "Natural convection from a discrete heat source on the bottom of a horizontal enclosure, "Int. J. Heat Mass Transfer, vol.43, pp.no 2257–2266,2000.
[7] H.Y. Wang, F. Penot, J.B. Saulnier, "Numerical study of a buoyancy-induced flow along a vertical plate with discretely heated integrated circuit packages, " Int. J. HeatMass Transfer ,vol.40 ,1997,pp.no 1509–1520.
[8] Y. Liu, N. Phan-Thien, "An optimum spacing problem for three chips mounted on a vertical substrate in an enclosure, " Numer. Heat Transfer, Part A ,vol.37, pp.no 613–630,2000.
[9] S. Chen, Y. Liu, "An optimum spacing problem for three-by-three heated elements mounted on a substrate, "Heat Mass Transfer,vol. 39,2000.
[10] I. Sezai, A.A. Mohamad, "Natural convection from a discrete heat source on the bottom of a horizontal enclosure, "Int. J. Heat Mass Transfer ,vol.43, pp.no 2257–2266,2000.
[11] T.J. Heindel, F.P. Incropera, S. Ramadhyani, "Laminar natural convection in a discretely heated cavity. 2. Comparisons of experimental and theoretical results, " J. Heat Transfer,vol.117,pp.no 910–917,1995.
[12] T.J. Heindel, S. Ramadhyani, F.P. Incropera, "Laminar natural convection in a discretely heated cavity. 1. Assessment of three-dimensional effects, "J. Heat Transfer,vol. 117,1996.
[13] T.J. Heindel, F.P. Incropera, S. Ramadhyani, "Laminar natural convection in a discretely heated cavity. 2. Comparisons of experimental and theoretical results, " J. Heat Transfer,vol. 117,pp.no 910–917,1995.
[14] A.K. da Silva, S. Lorente, A. Bejan, "Optimal distribution of discrete heat sources on a wall with natural convection, " Int. J. Heat Mass Transfer, vol.4,pp.no 203–214,2004.
[15] Y. Liu, N. Phan-Thien, "An optimum spacing problem for three chipsmounted on a vertical substrate in an enclosure, Numer. Heat Transfer, " Part A ,vol.37, pp.no 613–630,2000.
[16] A.K. da Silva, S. Lorente, A. Bejan, "Optimal distribution of discrete heat sources on a wall with natural convection, "Int. J. Heat Mass Transfer,vol. 47,pp.no 203–214,2004.
[17] I. Dagtekin, H.F. Oztop, "Natural convection heat transfer by heated partitions, "Int. Commun. Heat Mass Trans. ,vol.28,pp.no 823–834,2001.
[18] A. Merirkh, J.L. Lage, "Natural convection in an enclosure with disconnected and conducting solid blocks, "Int. J. Heat Mass Trans.,vol. 48,pp.no 1361–1372,2005.
[19] C.P. Desai, K. Vafai, M. Keyhani, "On the natural convection in a cavity with a cooled top wall and multiple protruding heaters, "J. Electron. Packaging,vol. 117, pp.no 34–45,1995.
[20] Fluent Inc,1995, "Computational Fluid Dymanics," Software,Lebanon, NH.
[21]Massimo Corcione, "Heat transfer correlations for free convections from upward-facing horizontal rectangular surfaces,"5th WSEAS Int. Conf. on Heat and Mass transfer (HMT'08),Vol 2,2007.
[22]Rajkumar.M.R, Venugopal.G & Anil Lal.S,P "Natural convection with surface radiation from a planar heat generating element mounted freely in a vertical channel," Heat Mass Transfer ,Vol. 47,pp. 789–805,2011.