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Simulation of Natural-Convection in Air-Cooling Electronic Components Mounted in a Horizontal Channel

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Abstract—In this investigation due to advancement in high–speed computers and different electronic components, heat transfer rate at the microchip is becoming a source of concern. Cooling is necessary for increasing the reliability and functionality of electronic components. Cooling of 2 to 5 identical heat sources mounted in a two-dimensional horizontal channel by natural convection air cooling is shown in present work. ANSYS software which is based on finite volume method uses Semi-Implicit Method for Pressure Linked Equations algorithm with second order upwind scheme for fully–developed flow to solve the conservation equations of mass, momentum and energy. Result shows that for each case heat transfer rate for first component increases then decreases for middle components and for last one it increases for Pr= 0.71 at Ri = 400.The Richardson number (Ri) represents the relative magnitude of natural to forced convection for a fluid. The range in which mixed forced and natural convection exit is 0.1<Ri<10. The temperature contours for 2 to 5 identical heat sources is shown in the study.

Index— Natural convection; ANSYS; Semi-Implicit; Richardson number; Prandtl number

1 INTRODUCTION

Due to advancement in high–speed computers and different electronic components, heat transfer rate at the microchip is becoming a source of concern. Until 1950 vacuum tubes played an important role for functioning of TV, radio, radar, and the digital computer. The best known computer in that era was ENIAC (ELECTRONIC NUMERICAL INTEGRATOR AND COMPUTER) in 1946. In 1948 marked the beginning of bipolar transistor. In 1959, with the introduction of (IC), where several components such as diodes, transistors, resistors and capacitors are packed in single electronic chip. The number of component increases drastically. According to Moor's law the number of transistors in per square inch in every two years would double, this creates a force in thermal engineers to focus on thermal management of electronics component by doing research on cooling of electronic components by different methods. Continuously decrement of electronic system resulted in heat generation per unit volume. The continuously decrement leads to medium-scale integration (MSI) with 50 to 1000 components per chip in 1960, large-scale integration (LSI) with 1000 to 100,000 components per chip in 1970, and very large-scale integration (VLSI) with 100,000 to 10,000,000 components per chip in 1980 **(Cengel)**. The current methods convection and radiation limits the heat transfer rate in mobile chip set, up to 1-2 W depending on size**(Ronan Grimes, Ed Walsh, Pat Walsh, 2010)** .As circuit density continuously increasing on silicon chip, power density rises, while chip temperature should be maintained below 85℃ (**R.K.Ali, 2009**). Currently there are five cooling strategies available *i.e.* (1)Free Convection (2) Forced Convection (3)liquid cooling (4)edge cooling (5) phase change cooling**(Jundika C. Kurnia,2010).**

In order to get an idea for thermal design of electronic system, the present study focus on effect of identical heat sources at Re = 5 is evaluated. In this paper, two dimensional transient laminar air flows in a horizontal channel with four cases is evaluated.

Nomenclature

g	gravitational acceleration, m/sec^2 T = temperature in K
Gr	Grashof number, $= g\beta(T_s - T_\infty)H^3/\vartheta_{air}^2$
H	height of the component, m
k	thermalconductivity of the fluid, W/m-K
k^*	dimensionless thermal conductivity, k/k_{air}
$\overline{Nu}$	Average Nusselt number $= \frac{\bar{h}H}{K}$
$\bar{h}$	Average heat transfer coefficient
Re	Reynolds number $= \frac{U_0 H}{\vartheta_{air}}$
Pr	Prandtl number, $= \frac{\vartheta_{air}}{\alpha_{air}}$
Ra	Rayleigh number = Gr.Pr
Ri	Richardson number $\frac{Gr}{Re^2}$
U_0	uniform inlet velocity, m/s
U, V	dimensionless horizontal and vertical Cartesian coordinates,=(x,y)/H
θ	dimensionless temperature $= \frac{T - T_0}{T_s - T_0}$
α	thermal diffusivity of the fluid, m^2/sec
β	thermal expansion coefficient of the fluid $\frac{1}{k}$
τ	dimensionless time, $= t/(H/U_0)$

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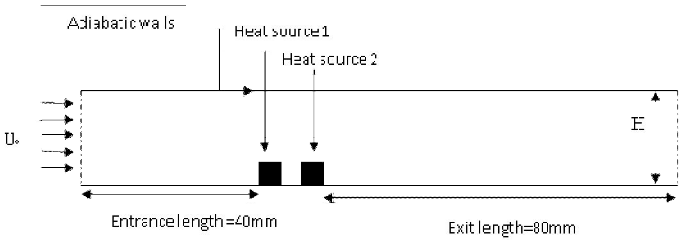
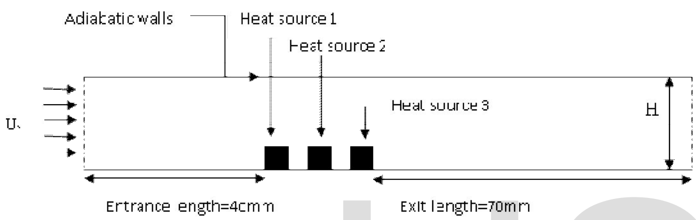
2. MODELLING OF GEOMETRY

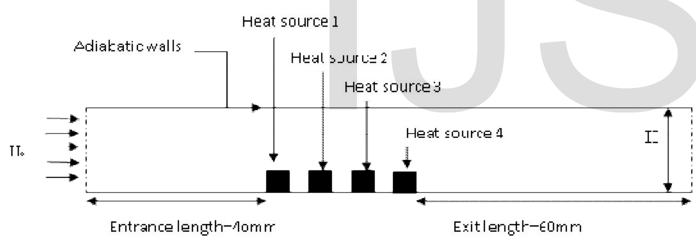
In the study, in a horizontal channel of height (H) and length of 20 and 135mm, identical heat sources of each size of 5mm×5mm are mounted in all four cases. The entrance length in all cases is same i.e. 40mm and distance between heat sources is same in all cases i.e. 5mm. The figures of all cases are shown in below.



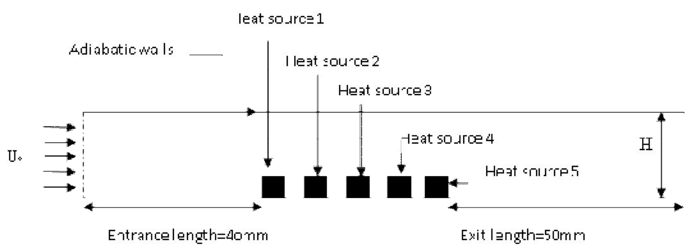
(a)



(b)



(c)



(d)

Fig.2 Schematic diagram from 2 to 5 heat sources in (a), (b), (c) and (d)

The top and bottom walls of a channel for each case are adiabatic. An air with a velocity of $U_\circ$ enters with a uniform temperature of $T_\circ$ inside a channel in all four cases. The heat sources are kept at uniform temperature of T_s.

In dimensionless form all conservation equation of mass, momentum and energy for all cases for two dimensional laminar natural convection and incompressible flow with all constant properties and with Boussinesq approximation can be written as shown below

$$\frac{\partial U}{\partial X}+\frac{\partial U}{\partial Y}=0 \qquad (1)$$

$$\frac{\partial U}{\partial \tau}+U\frac{\partial U}{\partial X}+V\frac{\partial U}{\partial Y}=-\frac{\partial P}{\partial X}+\frac{1}{Re}\left\{\frac{\partial}{\partial X}\left(\vartheta^{*}\frac{\partial U}{\partial X}\right)+\frac{\partial}{\partial Y}\left(\vartheta^{*}\frac{\partial U}{\partial Y}\right)\right\} \qquad (2)$$

$$\frac{\partial V}{\partial \tau}+U\frac{\partial V}{\partial X}+V\frac{\partial V}{\partial Y}=-\frac{\partial P}{\partial Y}+\frac{1}{Re}\left\{\frac{\partial}{\partial X}\left(\vartheta^{*}\frac{\partial V}{\partial X}\right)+\frac{\partial}{\partial Y}\left(\vartheta^{*}\frac{\partial V}{\partial Y}\right)\right\}+\frac{Gr}{Re^{2}}\theta \qquad (3)$$

$$\frac{\partial \theta}{\partial \tau}+U\frac{\partial \theta}{\partial X}+V\frac{\partial \theta}{\partial Y}=-\frac{1}{Re\,Pr}\left\{\frac{\partial}{\partial X}\left(k^{*}\frac{\partial U}{\partial X}\right)+\frac{\partial}{\partial Y}\left(k^{*}\frac{\partial U}{\partial Y}\right)\right\} \qquad (4)$$

Where,

$\vartheta^{*}=\frac{\vartheta_s}{\vartheta_{air}}\rightarrow\infty$ in each heat source

$\rightarrow 1$ in the fluid region

$k^{*}=\frac{k_s}{k_{air}}\rightarrow\infty$ in each heat source

$\rightarrow 1$ in the fluid region

$\vartheta^{*}\rightarrow\infty$ In order to get no slip condition *i.e.* U = V = 0 in each level of heat source and $k^{*}\rightarrow\infty$ in order to get uniform temperature over the heat source .

Now, the dimensionless initial and boundary condition for all heat sources are shown below:

Initial condition
At $\tau = 0$, U= V= 0

The boundary condition for all cases for present study at $\tau > 0$ as shown below

At X = 0 and $0\le Y\le 1$; U=1 V= 0, $\theta = 0$ (inlet)

At X = L and $0\le Y\le 1$; $\frac{\partial U}{\partial X}$ V = 0, $\frac{\partial \theta}{\partial X} = 0$ (outlet)

At Y = 0 and $0\le X\le 1$; U= 0 V = 0, $\theta = 0$ (adiabatic wall)

At Y = 1 and $0\le Y\le L$; V = 0, $\theta = 0$ (adiabatic wall)

3. Numerical Procedure

The governing equations from equation 1 to 4 are solved by finite volume method in ANSYS. A fully implicit time marching scheme is used. In solution method in ANSYS, SIMPLE algorithm for managing pressure-velocity coupling is used and transition k- kl-omega model is used. The temperature contour for two heat source in this study is shown in fig.3.1. The numerical code for two heat sources are validated with the numerical results of **Lyes Boutina and RachidBessaih (2009)** obtained the temperature contour for two heat source as shown in fig.3.2

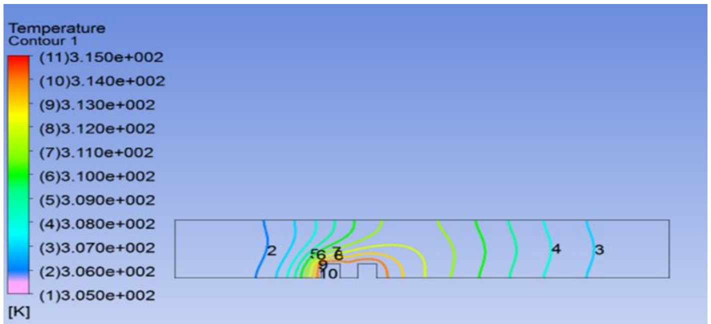


Fig. 3.1 Temperature contour for present study for Re =5 for case 1 at Gr = 10^4

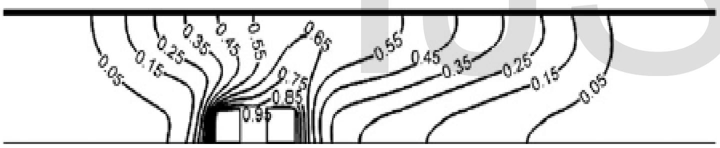


Fig. 3.2 Temperature contour for Adel Hamouche and Rachid Bessaih for Re = 5 at Gr = 10^4

Table 3.1 Validation of present work with Hamouche and Rachid Bessaih at Re=5

	Number of nodes	Nu_1	Nu_2	% error
Hamouche and Bessaih	19840	1.3806	0.7763	12.62%
Present study	10978	1.2063	0.6065	

4. Results and Discussions

The study deals with the number of heat sources mounted on the bottom wall at Re = 5 for all four cases. The temperature difference kept is 10K. The fluid inside the channel is air and heat source is made up of aluminium. The inlet temperature is 305K and 315K heat source temperature.

Table 4.1 Thermal properties of air

Fluid	ρ_f (kg/m^3)	k_f (W/m-K)	μ_f (kg/m-s)	c_p(J/kg-K)	T(K)
Air	1.1717	0.02624	1.725×10^{-5}	1004.9	305

Table 4.2 Thermal properties of solid

Solid	ρ_s (kg/m^3)	k_s (W/m-K)	c_p(J/kg-K)
Aluminium	**2719**	**202.4**	**871**

Table 4.2 Effects of Number of Heat sources

No. of heat sources	Nu_1	Nu_2	Nu_3	Nu_4	Nu_5
2	1.206	0.606	-	-	-
3	1.7824	0.5882	0.61312	-	-
4	1.7	0.56084153	0.350836	0.45235	-
5	1.77662	0.57515	0.350909718	0.231517	0.3578

From Table 4.2 it can be noticed that in each case from 3 to 5 heat sources, the average nusselt number of last one is always greater than the second last, it is because of high thermal mixing of fluid is possible for last one and which is due to more-free space for last heat source.

The temperature contours for last three cases is shown in fig.4.1, fig. 4.2, and fig. 4.3. All the contours are slightly same for all four cases.

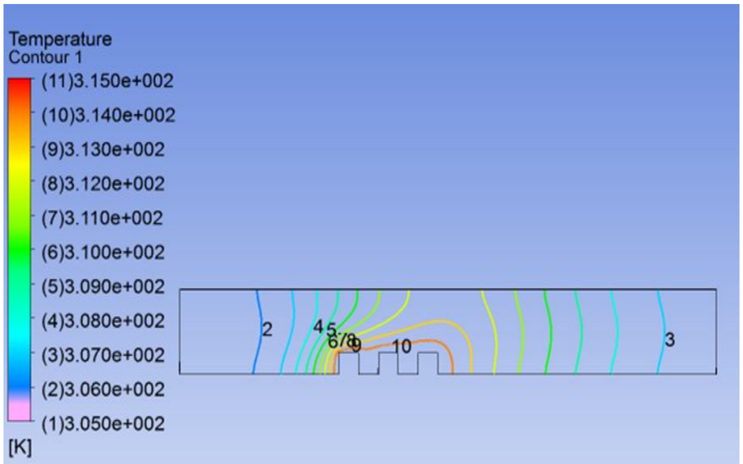


Fig.4.1Temperature contour for 3 heat sources at Re = 5at Gr = 10^4

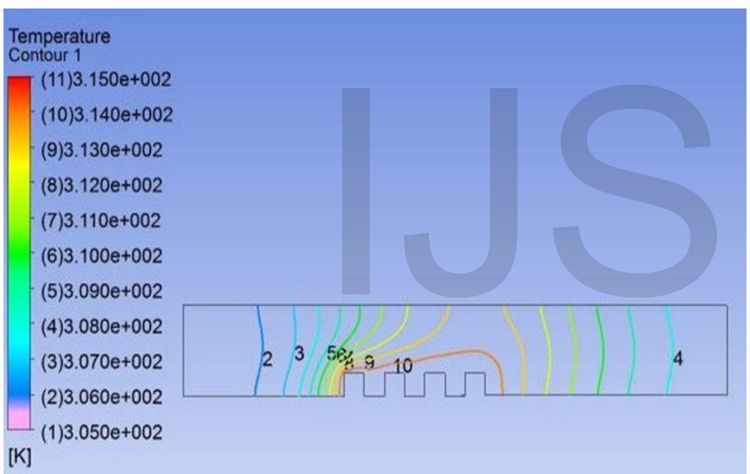


Fig.4.2 Temperature contour for 4 heat sources at Re = 5 at Gr = 10^4

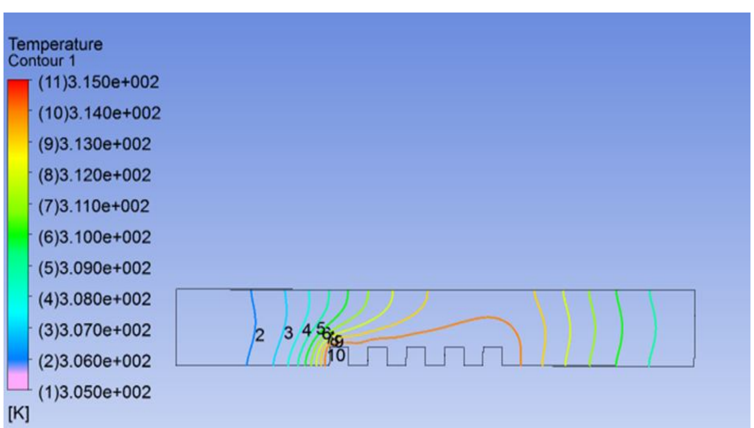


Fig.4.3 Temperature contour for 5 heat sources at Re = 5 at Gr = 10^4

5. Conclusion

In this study from 2 to 5 heat sources the average nusselt number of last one is always greater than the second last, it is because of high thermal mixing of fluid is possible for last one and which is due to more free space for last heat source. The percentage error in average nusselt number in the study as compare to Adel hamouch and Rachid Bessaih (2009) is 12.62%. The temperature profile for all four cases is slightly same. The entire temperature contour generated in ANSYS.

6. References

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